

# HIGH-EFFICIENCY, ULTRA-LOW OXIDES OF NITROGEN SUPPLEMENTAL FIRING BURNER

*Prepared For:*

**California Energy Commission**  
Public Interest Energy Research Program

*Prepared By:*



Arnold Schwarzenegger  
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## Preface

The California Energy Commission's Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

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- Transportation

*High Efficiency Ultra-Low NO<sub>x</sub> Supplemental Firing Burner* is the final report for the Combined Heat and Power project (Contract Number 500-03-040) conducted by ALZETA Corporation. The information from this project contributes to PIER's Environmentally Preferred Advanced Generation Program.

For more information about the PIER Program, please visit the Energy Commission's website at [www.energy.ca.gov/research/](http://www.energy.ca.gov/research/) or contact the Energy Commission at 916-654-4878.



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## Abstract

This project developed and tested a high-efficiency, ultra-low oxides of nitrogen (NO<sub>x</sub>) supplemental firing burner system. The target market is California, where ultra-low NO<sub>x</sub> regulations limit the use of conventional supplemental firing burners in combined heat and power facilities. Activities included a prototype phase, which provided baseline design data and developed a method for predicting NO<sub>x</sub> emissions; a pilot phase, in which a burner module that would be appropriate for industrial applications was developed and tested; and a full-scale testing phase, where a number of burner modules were installed in an operating combined heat and power facility. Results showed that fewer than 3 parts per million NO<sub>x</sub> (corrected to 15 percent oxygen) could be achieved in the turbine exhaust gas environment typical of many combined heat and power facilities. However, an initial full-scale test in a turbine exhaust gas environment failed structurally, as the turbulence and flow variations present in the exhaust stream were much greater than anticipated. A second test with a more rugged burner system design was structurally successful, but ultra-low NO<sub>x</sub> emissions were not achieved due to of the turbine exhaust gas turbulence. This turbulence significantly decreases gas flow uniformity, which is critical to optimizing performance and minimizing emissions. A commercial system will need to address the turbine exhaust gas turbulence and flow variations to ensure ultra-low NO<sub>x</sub> emissions.

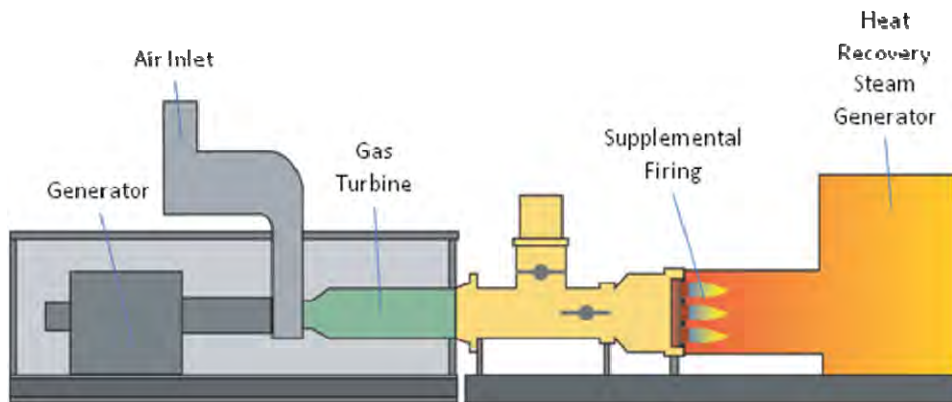
**Keywords:** Combined heat and power, CHP, supplemental firing burner system, ultra-low NO<sub>x</sub> emissions



## Executive Summary

The primary goal of this project was the development of a supplemental firing burner system (see Figure 1) to improve the efficiency of combined heat and power facilities while providing the end user with ultra-low emissions of oxides of nitrogen (NO<sub>x</sub>). By reheating turbine exhaust gas, overall plant efficiency is increased, and steam generation in the heat recovery steam generator can match demand independently from the operation of the gas turbine. The NO<sub>x</sub> emissions goal is 3 parts per million corrected to 15 percent oxygen, which is essential for the California market. The burner and controls technology developed by this contract also would have broader application, in that ultra-low NO<sub>x</sub> technology is desired in process industries where supplemental firing is used in providing hot air for numerous drying applications in the food and plastic industries, among others.

Turbine exhaust gas is the hot flow downstream of a turbine. It typically is hot (between 700°F and 1200°F) with reduced oxygen content relative to ambient air (15 percent oxygen to 18 percent oxygen in turbine exhaust gas versus 21 percent oxygen in air). The turbine exhaust gas flow is also a relatively high velocity and highly turbulent stream confined in the duct connecting the turbine exhaust to the inlet of the heat recovery steam generator. The current state-of-the-art supplemental firing burners, or “duct burners,” are modules that do not provide any, or very limited, fuel and air premixing. Essentially, fuel is injected into the turbine exhaust gas stream, and the design of the duct burner “holds” the flame in one place with the flame extending 5 to 10 feet or more down the length of the duct. This “diffusion flame” combustion produces NO<sub>x</sub> emissions well in excess of the emissions requirements in many areas within California.



**Figure 1. Typical cogeneration system**

Source: <http://www.eclipsenet.com/catalog/contents/Documents/01/165/165C%20Bulletin.pdf>

As an alternate approach, the high-efficiency, ultra-low NO<sub>x</sub> supplemental firing burner technology fully premixes the fuel and air. Combustion takes place on a metal mesh-like surface, with flames extending only inches off the surface. A system is made up of modules, each with a capacity of 1.2 million British thermal units per hour. The module itself is a box

structure, open at one end, and having a wedge-shaped mesh surface at the other. Turbine exhaust gas flows into the open end of the box where it is mixed with fuel at an optimum air-to-fuel ratio to promote low NO<sub>x</sub>. This premixed fuel and turbine exhaust gas continues to flow through the box and out through the wedge-shaped mesh surface. Combustion takes place on the outermost surface of the mesh, and because turbine exhaust gas and fuel has been optimally mixed, NO<sub>x</sub> emissions are very low.

This project was divided into three technical phases. In the prototype phase, the turbine exhaust gas environment was simulated at laboratory scale (typically 800°F, and as low as 15 percent oxygen), premixed burners were tested in this environment, and NO<sub>x</sub> emissions were measured. It was demonstrated that NO<sub>x</sub> emissions can be correlated to *adiabatic flame temperature*. The adiabatic flame temperature is the temperature that would be reached by a fuel-oxidizer mixture if it could be completely combusted in an environment with no heat loss. Establishing this relationship between adiabatic flame temperature and NO<sub>x</sub> emissions means performance can be predicted based on turbine exhaust gas conditions entering the supplemental firing burner. These results of the prototype tests justified proceeding to the pilot scale phase.

In the pilot scale phase, a modular design was developed that achieved a better than 2:1 burner surface-to-duct cross sectional area ratio. This ratio is necessary to match the typical heat flux (the rate of heat energy transfer through a given surface) entering heat recovery steam generators without expanding the duct size. The ability to meet performance goals without requiring changes to existing plant hardware is essential for retrofit applications. It was also shown that the emissions and pressure drop relationship identified during the prototype phase would scale directly with the burner surface area.

The ultra-low supplemental firing burner technology was then subjected to a full-scale field test at an existing combined heat and power facility. Conditions were similar to the pilot phase in exhaust flow velocity and oxygen concentration, but the gas temperature was 1050°F (about 250°F higher). Only hours after installing the 10-burner module system, a structural failure occurred, and testing in the actual turbine exhaust gas environment could not take place. It is believed that the high turbulent environment and vibrations from equipment subjected the burner modules to excessive stresses.

As a result of the structural failure, the module design was analyzed by a structural engineering company. The engineers recommended several improvements, including increasing the size of the burner support tube to minimize deflection and resist vibration, adding additional supports at the rear of the module to reduce flutter, and altering the design slightly to improve weld penetration at the joints. A more robust, multimodule design was then fabricated and tested, allowing combustion performance to be evaluated. While the redesign proved structurally sound, these results showed that uniform combustion is very difficult to achieve in the turbine exhaust gas environment and, therefore, emissions goals were not achieved. Based on pilot phase results, the ultra-low NO<sub>x</sub> supplemental firing burner technology would be successful if the flow entering the burner module could be more uniform.

In spite of the difficulties that were encountered during the project, the potential benefits of this ultra-low NO<sub>x</sub> technology are significant and merit further study. For a current generation combined heat and power system to be installed in the greater Los Angeles metropolitan area, the San Joaquin Valley, or the Bay Area, the complete system (including the power generating system and the duct burner) must meet NO<sub>x</sub> emissions levels that exceed what can be achieved with current burners. Stack treatment systems, such as selective catalytic reduction, provide a solution, but the solution is much more expensive than a burner solution. The result is a higher cost for power and heat delivered by the combined heat and power system, which makes the combined heat and power system less cost-effective and ultimately limits the use of combined heat and power in California. The availability of a viable ultra-low NO<sub>x</sub> duct burner and gas turbine combustor would benefit California's ratepayers by reducing the overall cost of combined heat and power.

The following recommendations would address some of the problems that existed at the end of this project. First, a characterization of actual turbine exhaust gas environments is required, including flow variations and turbulence. With a better understanding of the turbine exhaust gas environment, the hope is to identify ways of making the flow more uniform so that combustion stability could be improved. Second, an alternate approach should be evaluated that withdraws turbine exhaust gas from the duct, premixes it with fuel, and then reintroduces the premix to burners inside of the duct. This approach could virtually eliminate the need to deal with turbine exhaust gas flow variations and turbulence.

The data in the figures and tables of this document was obtained by the author unless otherwise noted in a source line beneath the table in question.



## 1.0 Introduction

This project's goal was to develop and demonstrate an ultra-low oxides of nitrogen (NO<sub>x</sub>) supplemental firing burner technology for improving the efficiency of industrial scale combined heat and power (CHP) generating systems. Improvements were to be achieved by: a) providing very high overall thermal efficiency, with the efficiency target being to meet or exceed the efficiency of state-of-the-art industrial boilers; b) improving the operational flexibility with respect to the balancing heat and power loads to better match process needs; and c) having very low NO<sub>x</sub>, carbon monoxide (CO), and hydrocarbons emissions, also to be consistent with the state-of-the art industrial boiler burners.

The desired burner technology also would be flexible enough in its design and operation to be applicable to a wide range of CHP systems. Yet, for the purpose of completing this project, the development had to be focused on a specific application. After reviewing several approaches, a supplemental firing burner system for reheating the exhaust from an industrial gas turbine-generator was selected, such that the burner would be installed between a turbine and a heat recovery steam generator (HRSG). Although this is a common application for supplemental firing, the approach is also applicable to the exhaust from reciprocating engines and some fuel cell technologies, recirculating hot air ducts, or any process needing downstream exhaust heating.

### 1.1. Project Background

The specific application will target a "typical" CHP customer with on site power requirements in the capacity range of approximately 1 MW<sub>e</sub> to 5 MW<sub>e</sub>. There are a number of gas turbine manufacturers in this capacity range. Two examples of manufacturers of this class of equipment are Solar Turbines of San Diego, California, and Kawasaki Gas Turbines – Americas of Grand Rapids, Michigan. A comparison of published performance data for representative turbines from these manufacturers is presented in Table 1.

A review of Table 1 presents the following relevant information:

- Industrial gas turbine efficiencies are relatively low—in the 20-30% range based on fuel HHV – and therefore generate a large amount of waste heat relative to the amount of power produced.
- The waste heat that is produced is a high volume of relatively low temperature gas, which limits the effectiveness of the heat recovery that can be achieved unless supplemental firing is included.
- For a typical industrial user, the cost of fuel alone can be 80% or more of the avoided cost of purchased electricity if there is no downstream heat recovery. When other costs such as amortized capital costs and non-fuel operating and maintenance costs are included, generation of electricity without downstream heat recovery is rarely economical.
- There can be reasons other than cost savings to produce power on site, such as improved system reliability or to overcome the inability to get sufficient grid power to a specific site. However, in general the use of gas turbines to provide distributed

generation requires some form of waste heat recovery in order to be economically feasible. Improved cost-effectiveness of waste heat recovery can dramatically improve the cost-effectiveness of CHP systems.

**Table 1. Performance characteristics of 1-5 MWe gas turbines**

<b>Turbine Type</b>	<b>Solar Saturn 20</b>	<b>Kawasaki GPB15DLE</b>	<b>Solar Taurus 60</b>
Output Power MWe @590F, (150C)	1.21	1.42	5.20
Heat Rate <sup>1</sup> Btu/kWe-hr (kJ/kWe-hr)	15,500 (16,400)	16,200 (17,100)	12,500 (13,200)
Efficiency <sup>1</sup> , % (based on fuel HHV)	22.0	21.1	27.3
Exhaust Flow, lb/hr (kg/hr)	51,240 (23,220)	63,300 (28,710)	174,800 (79,280)
Exhaust Temp, °F (°C)	960 (516)	995 (535)	906 (486)
Fuel Component of Power Cost, \$/kWe-hr (\$5/MMBtu fuel)	\$.078	\$.081	\$.063

1. Heat input is based on fuel HHV. Turbine industry typically uses LHV, which results in lower calculated heat rate and higher calculated efficiency.

### **1.1.1. Introduction of Technology**

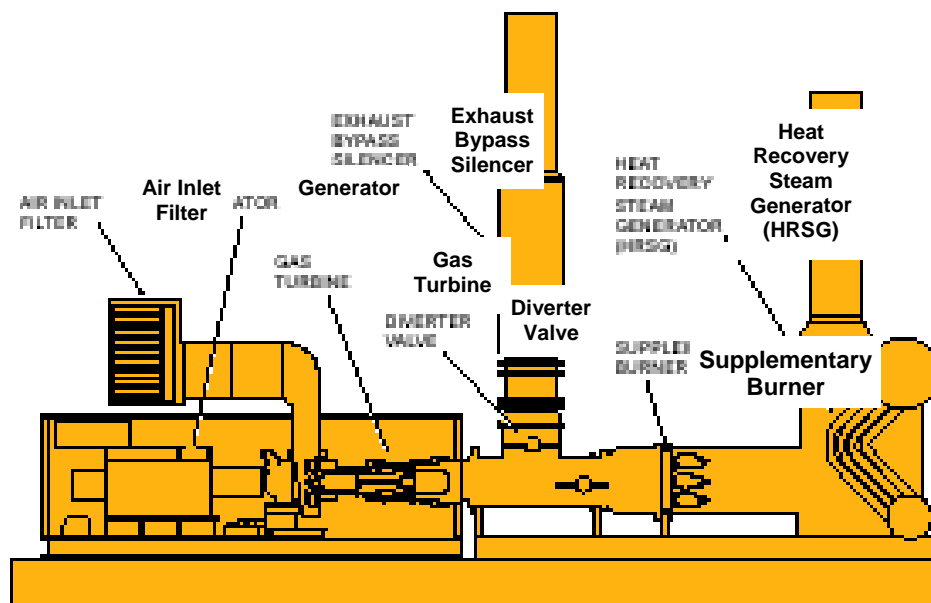
#### **Downstream Heat Recovery From Waste Heat Boilers and HRSGs**

In the 1-5 MWe size range, turbine exhaust heat recovery is commonly achieved by using a waste heat boiler or heat recovery steam generator (HRSG). In the simplest systems, there is no supplemental fuel added to the turbine exhaust. These unfired systems have the advantage of being very simple to operate and avoid the complexity of adding a second pollution source from the burners of the supplemental firing system. But there are significant disadvantages:

- Compared to a similarly sized industrial boiler, the thermal efficiency of the unfired HRSG is significantly lower. Useful energy is being “lost.”
- The amount of steam that is produced is nearly constant for a given gas turbine size, providing the facility operator with much less steam supply “turndown” than could be realized with a conventional industrial boiler.

Improved operating flexibility, higher thermal efficiency, and an increased steam-to-power ratio can all be obtained by installing supplemental duct burners between the turbine and the waste heat boiler or HRSG and heating the exhaust gases, as shown in Figure 2. These burners elevate the temperature of the turbine exhaust gas (TEG) to increase steam production and can provide more control over the amount of steam produced relative to the power produced by the gas turbine. Marginal efficiency (conversion of fuel energy to steam) is very high.

Examples of how duct burners and downstream heat recovery equipment would interface with a gas turbine from an energy balance perspective are provided in Table 2. These calculations were done using a Solar Turbines Saturn 20 engine as the power generator. Heat recovery values and efficiencies were calculated assuming a stack temperature for the combustion products of 300°F. On the steam side, 150 pounds per square inch gauge (psig) steam was assumed with an inlet water temperature of 227°F and exit conditions of 150 psig at saturated conditions. These inlet and exit steam conditions are considered to be typical for industrial steam applications. The stack temperature is considered to be achievable in systems designed for high thermal efficiency.



**Figure 2. Cogeneration system. Supplementary burners are located between turbine and heat recovery steam generator.**

Source: Solar Turbines

Although these calculations were done for the Saturn 20, similar heat recovery numbers could be calculated for other industrial turbines currently used for industrial power generation and cogeneration systems without significantly changing the conclusions that will be reached. For example, a 5 MW<sub>e</sub> turbine will have a TEG mass flow that is roughly 5 times the mass flow of a 1 MW<sub>e</sub> engine and would generate roughly 5 times the amount of steam in a HRSG fired at a similar temperature. After correcting for the difference in capacity, there will be small differences in mass flow and TEG temperature due to differences in efficiency, but as stated above, these differences are relatively small within the framework of this discussion of the benefits provided by supplemental firing.

Table 2 presents the following information:

- The first column provides data for the case with no downstream heat recovery. In industrial applications this approach would rarely be cost-effective but is included to show the efficiency of the turbine by itself with no heat recovery.

- Downstream heat recovery equipment adds backpressure to the turbine, which has a slight negative impact on the turbine efficiency. A power reduction of 0.25% per in. water column (w.c.) increase in back pressure is typical and is reflected in Table 2. A heat recovery boiler will add backpressure, and supplemental firing burners will add additional pressure drop above what has been added by the boiler. Therefore, in Table 2 approximately 6 in. w.c. was assumed and added in the unfired HRSG case; and 12 in. w.c. was added in the fired HRSG case. The resultant power loss is tabulated.

**Table 2. Comparison of heat recovery approaches including baseline and maximum heat recovery (1.2 MW<sub>e</sub> engine)**

<b>Heat Recovery Approach</b>	<b>No Heat Recovery</b>	<b>Unfired Heat Recovery; 6 in. w.c. backpressure</b>	<b>Supplemental Firing (1600°F); 12 in. w.c. backpressure</b>	<b>Optimum Supplemental Firing (3% Stack O<sub>2</sub>)</b>
Power Generation (MW <sub>e</sub> )	1.21	1.19	1.17	1.17
Turbine Heat Input (MMbtu/hr)	18.8	18.8	18.8	18.8
Energy in Turbine Exhaust, MMBtu/hr	13.8	13.8	13.9	13.9
Supplemental Fuel MMBtu/hr			11.3	43.3
Cogeneration Heat Recovery MMBtu/hr		8.9	18.8	46.9
Steam Generation (lb/hr)		8,900	18,800	46,900
Power-Steam Ratio (%) <sup>1</sup>		45.6%	21.2%	8.5%
Overall System Efficiency, HHV	22.0%	69.2%	75.7%	82.0%

1. Power-steam ratio is defined as the ratio of electric power to steam output in equivalent units of power (MW<sub>e</sub>/MW<sub>th</sub> for example).

- The second column “Unfired Heat Recovery” presents the case where there is downstream heat recovery without supplemental firing. The amount of steam generated by this approach provides a power-to-steam ratio of 0.46 in this specific case and would typically be between 0.4 and 0.5. There is very limited flexibility in the amount of steam that can be generated. The peak steam supply is fixed at 8,900 lb/hr. Consequently, the only way to reduce the amount of steam generation at a fixed turbine load is to vent the TEG to atmosphere, bypassing the steam generator and therefore “bypassing” any efficiency gain from the system.
- The column “Supplemental Firing (1600°F)” presents the performance of a typical duct burner using today’s baseline performance. The temperature rise across the burner would be 600-650°F. Steam production would be approximately doubled, from the unfired value of 8,900lb/hr to a fired value of 18,800 lb/hr. In addition, the system now

provides a variable steam supply of between 8,900 lb/hr and 18,800 lb/hr as the duct burner heat input is varied. Thermal efficiency would be lower at lower steam generating rates, matching the efficiency of the unfired heat recovery case at the 8,900 lb/hr condition and increasing to 75.7% efficiency at maximum steam rate.

- The last column “Optimum Supplemental Firing” represents a reasonable maximum thermal efficiency and maximum steam generation for a CHP system. By adding enough fuel to reduce the stack O<sub>2</sub> level to 3%, the thermal efficiency of the CHP system is now similar to the best thermal efficiency achieved by industrial boilers and process heaters. While this is the “best thermal efficiency,” it is also a condition that is routinely achieved in industrial boiler applications. This level of thermal efficiency is the reasonable maximum efficiency that could be achieved with an advanced duct burner concept in industrial steam applications.
- Also in the “Optimum Supplemental Firing” case, there is now a 5:1 steam turndown when operating between the unfired mode and the fully-fired case. By modulating the fuel input between these two limits, the steam supply from the HRSG becomes almost as flexible as that from a standard industrial boiler, where typical operational turndown ranges from 5:1 to 10:1.

### **1.1.2. Problem Statement**

#### ***Burners for Boilers Versus Burners for Supplemental Firing***

Today’s ultra-low NO<sub>x</sub> burners (ULNBs) for boilers have much lower NO<sub>x</sub> and CO emissions than duct burners. Sub-9 ppm NO<sub>x</sub> emissions and sub-50 ppm CO emissions (corrected to 3% O<sub>2</sub>) can be achieved by a number of manufacturers of burners for boilers, and there are hundreds of proven boiler applications in California. Conversely, typical “low NO<sub>x</sub>” duct burner emissions are on the order of 20-25 ppm NO<sub>x</sub> corrected to 15% O<sub>2</sub>, or 60-75 ppm NO<sub>x</sub> when corrected to the same 3% stack O<sub>2</sub> as for boiler applications. *The relatively high emissions from duct burners limits the use of CHP systems in California, Texas, and several other industrialized areas in the United States, and will continue to limit the economic viability of CHP systems if this problem is not addressed.*

Because the O<sub>2</sub> levels used as emission correction factors differ between boilers and gas turbines (3% for boilers and furnaces, which is considered a typical stack O<sub>2</sub> concentration; vs. 15% for turbines and ancillary components such as duct burners, which is also a typical stack O<sub>2</sub> concentration for these units), it may be simpler to reference emissions to lbs of NO<sub>x</sub> generated per MMBtu heat input. A lb-per-MMBtu comparison yields the same numerical value for either technology. The level that corresponds to 9 ppm NO<sub>x</sub> at 3% O<sub>2</sub> (and 3 ppm NO<sub>x</sub> at 15% O<sub>2</sub>) is .01 lb per MMBtu. Hence, *the goal of the project was this .01 lb per MMBtu level.* Table 3 presents representative baseline performance parameters for a current generation duct burner. This is compared to what is considered technically achievable.

The difference in the emissions of today’s best boiler burners versus today’s best duct burners raises the following questions:

- How do boiler burners achieve ULN emissions?

- Why can't duct burners reach the same emissions level?

In answering the first question, the generally accepted method for reducing NO<sub>x</sub> emissions in boilers and most large industrial burners is to reduce the flame temperature by adding additional air or flue gas. Since NO<sub>x</sub> formation varies exponentially with flame temperature, higher levels of dilution lead to more significant levels of NO<sub>x</sub> reduction. The use of flue gas is preferred over the use of excess air for thermal efficiency reasons, even though more fan power may be required to move flue gas through a burner.

**Table 3. Current baseline for supplemental firing burners and boiler burners**

Parameter	Duct Burner Baseline	Boiler Baseline	Comments
NO <sub>x</sub> Emissions (lb per MMBtu)	.06	.01	Boiler baseline corresponds to 9 ppm NO <sub>x</sub> at 3% stack O <sub>2</sub> which is widely regarded as boiler BACT. Duct burner emitting at .06 lb/MMBtu requires stack treatment to meet BACT.
CHP Thermal Efficiency	75.7%	82.0%	Baseline duct burner efficiency limited by low temperature rise across burner and high residual oxygen in exhaust.
Steam turndown	2.1:1	5.3:1	Limited turndown from current baseline limits operational flexibility. A minimum steam turndown of 5:1 is the expectation of industrial boiler operators.

Currently, the most successful manufacturers of ULNBs all operate at similar levels of excess combustion air and flue gas recirculation (FGR). These manufacturers include ALZETA and Todd Combustion and Coen (both John Zink Companies). Since the fundamental principle involved in reducing NO<sub>x</sub> is to lower flame temperature, it is not surprising that similar air and FGR levels are required. Nominal air and flue gas flows that are required are 20% excess combustion air and 35% FGR to achieve 9 ppm. ULNBs can run at FGR levels of 35-40%, resulting in O<sub>2</sub> levels in the air-flue gas mixture of 15-16%.

The second question posed above (Why can't duct burners achieve the same emissions levels?) is more difficult to answer. However, a review of available supplemental firing technologies confirms that current duct burners do not meet the 9 ppm NO<sub>x</sub> emissions level corrected to 3% stack O<sub>2</sub>. There are a number of reasons for this. The most important ones are:

- To minimize pressure losses downstream of the turbine, conventional duct burner technology is installed as an array (or grid) of burners that do not completely block the cross-section of the exhaust duct. This approach has the advantage of keeping pressure losses low, but it limits the amount of turbine exhaust that can interact with supplemental fuel, since all of the added fuel can not immediately mix and burn with all of the TEG. Therefore, for a given heat input, the flame temperature in the immediate vicinity of the fuel injection point is higher than the temperature that would

be reached in a premixed or rapidly mixed flame. This higher temperature leads to higher NO<sub>x</sub> emissions.

- In designing duct burners, the performance goals of low pressure drop and short flame length are in general not compatible with achieving the low peak flame temperatures that can be achieved in ULNBs designed specifically to operate in large furnaces. When power generation is the primary motivation for installing the system, the effectiveness of the heat recovery system may suffer.
- Although TEG has an O<sub>2</sub> level consistent with the air-FGR mixture of a ULNB, the turbine exhaust is typically at a temperature of 950-1000°F, which is significantly higher than the typical ULNB oxidizer temperature of 200°F. With the higher starting temperature for the combustion reaction, the resultant flame temperature is higher, resulting in higher NO<sub>x</sub> emissions.

## 1.2. Project Goals

One goal of this project was to close the technical gap that currently exists between duct burners and boiler burners with respect to both thermal efficiency and pollutant emissions. The project would build on technology that has been proven to work in other industrial applications and make a step change in the performance of supplemental firing technology.

Another issue is operational flexibility. Most industrial operators want boiler steam turndown capability of 10:1, with 5:1 turndown being a minimum expectation. An inability to achieve this full level of turndown can result in excess steam being vented to the atmosphere, an obvious loss of plant efficiency. If the steam generator cannot match plant requirements to steam supply, then either thermal efficiency suffers or operational complexity increases. The typical industrial gas turbine has a much more limited operating range with respect to mass flow than does a boiler. Therefore, it is difficult, if not impossible, to simultaneously meet plant electricity and steam demand with conventional cogeneration systems without suffering significant efficiency penalties during off-design operation.

For this reason, many plant operators have found it necessary to keep a redundant source of steam at the plant to provide steam when the cogeneration system is down and to increase operational flexibility with respect to boiler turndown. The cost of this redundancy is high. Assuming a cogeneration plant with an electric capacity of 5 MW<sub>e</sub> and a maximum steam requirement of 100,000 lbs/hr, the cost of a redundant steam supply can add 20 percent to the cost of the plant. Instead, these components could be replaced with a duct burner/HRSG, which could be fired separately when necessary.

As discussed in a technical paper “Comparative Evaluation of Cogeneration Cycle Alternatives” by R.C. Vetterick and P.G. Whitten (Reference 1), sizing of the electric and steam components of a cogeneration system should be driven by plant thermal requirements during normal operation. Equipment sizing should not be driven by maximum steam demand or maximum electric power usage due to inefficiencies that result from off-design performance. A plant operator can typically buy electricity off of the power grid but has no secondary source for steam. Generating more power than can be used on site, with the intent to sell

power back to the grid, is marginally feasible at best for industrial facilities in today's energy market.<sup>3</sup>

In order for cogeneration to become a more attractive and cost-effective option for industrial plant operators, systems must be developed that provide:

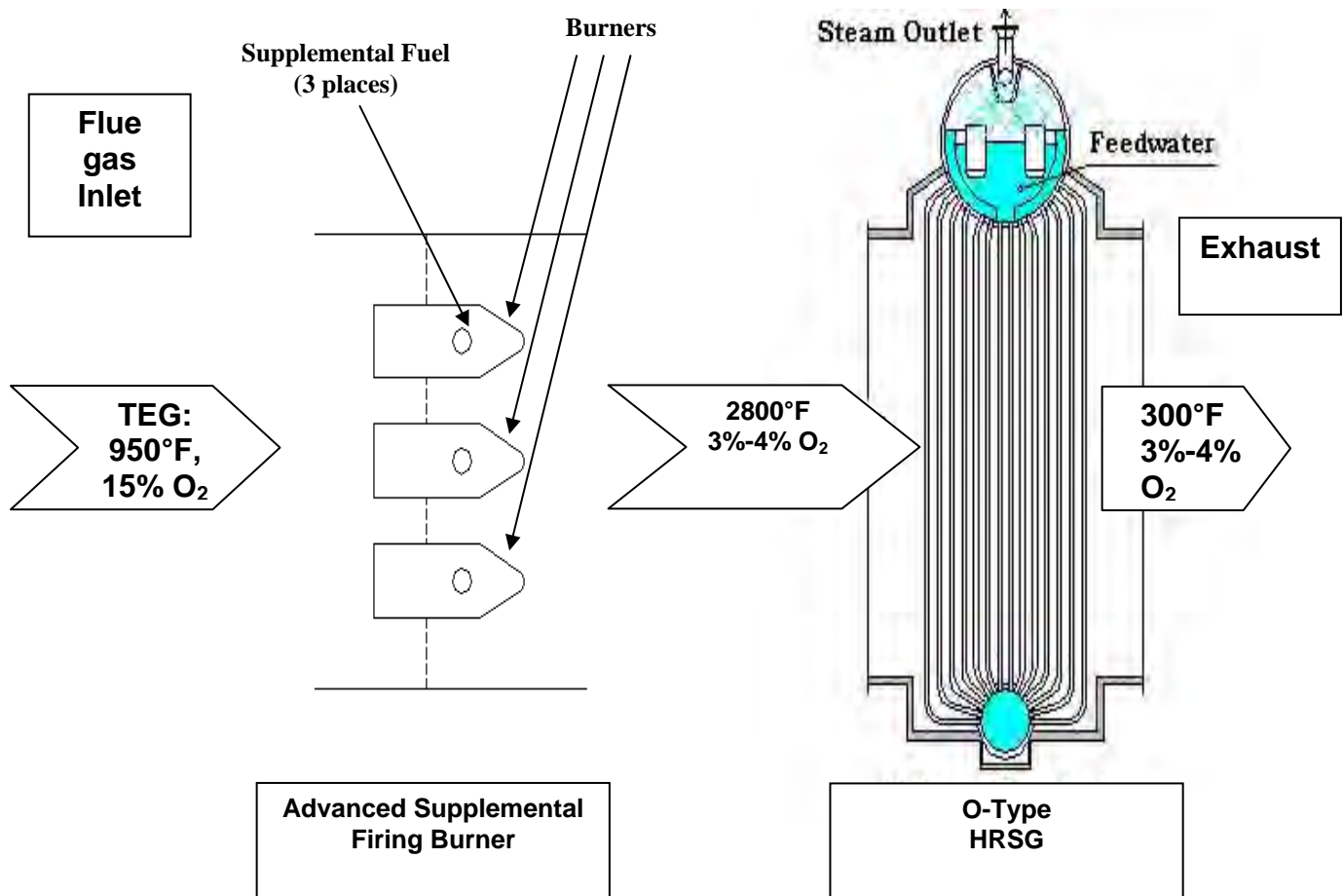
- High thermal efficiency.
- Operational flexibility of 5:1 minimum steam turndown, regardless of turbine operating condition.
- NO<sub>x</sub> and CO emissions that meet BACT cost-effectively and without stack treatment.

#### **1.2.1. Technical**

The technical basis for this work was to take proven fully premixed surface stabilized burner technology for boiler applications and extend this technology to operation in a turbine exhaust gas stream. A generic view of what the final product could look like is presented in Figure 3. As shown, the technology would be mounted in a duct to interface with both the upstream turbine exhaust and the downstream HRSG. Fuel is injected upstream of the burner surface. Note that the fuel is injected into discrete ports, with each gas injection port separately premixed with a portion of the flow.

This type of mixing is necessary due to the limited flammability of the oxygen depleted TEG stream. There is a fairly narrow window of flame stability with respect to fuel air ratio. Since the mass flow of turbine exhaust is essentially constant and the fuel-air operating range is narrow, significant turndown can be achieved only by separately staging each burner's operation. This same approach is used in current grid-type burner layouts, with the primary difference being that a design based on ULNBs requires fully premixing upstream, prior to combustion. One challenge in the duct burner application will be to stabilize combustion on the burner surface at the relatively high velocity that will exist in the surrounding duct due to both the limited duct cross-sectional area and the high temperature of the TEG.

While a larger duct would be preferred to allow a more planar burner arrangement, the CHP industry typically operates at a higher energy density. Therefore, a constraint is having to operate in smaller ducts. One potential approach is to install a burner geometry that has a surface area that is significantly larger than the projected area of the duct. Other options include cylindrical or conical burners mounted with their bases normal to the flow of the TEG. This trade-off of burner design complexity versus burner pressure drop and burner stability will be investigated in the project.



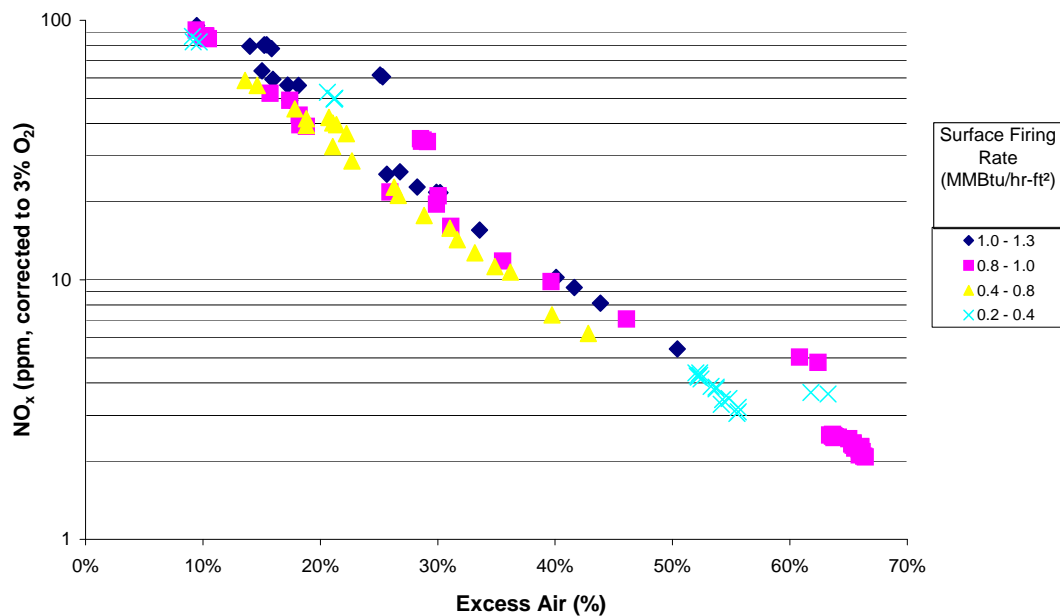
**Figure 3. Schematic of premixed surface stabilized supplemental firing burner with HRSG.**

The premixed surface stabilized technology that forms the basis of this duct burner is closely related to ALZETA's CSB commercial and industrial burner technology. ALZETA has been selling the CSB™ burner for 15+ years as a high excess air 9-ppm ULNB. The simplicity of the relationship between excess air, flame temperature, and NO<sub>x</sub> emissions can be easily demonstrated with this burner. Figure 3 shows this relationship. As excess combustion air is increased, NO<sub>x</sub> emissions decrease. Since NO<sub>x</sub> formation increases approximately as an exponential function of temperature, the relationship between NO<sub>x</sub> and excess air is nearly linear on a logarithmic plot of NO<sub>x</sub> emissions versus excess combustion air.

Operation at high excess air can reach sub-9 ppm emissions levels but with the penalty of lower thermal efficiency. A similar approach for lowering the flame temperature that minimizes the efficiency loss is to bring flue gas from the boiler stack back through the burner to cool the flame. This approach is known as flue gas recirculation (FGR). The FGR approach requires a similar level of "total dilution" of the flame to achieve the required NO<sub>x</sub> emissions level. In Figure 4, NO<sub>x</sub> emissions from a burner using different combinations of excess air and FGR are plotted versus "total dilution." Total dilution is defined as the total mass flow of

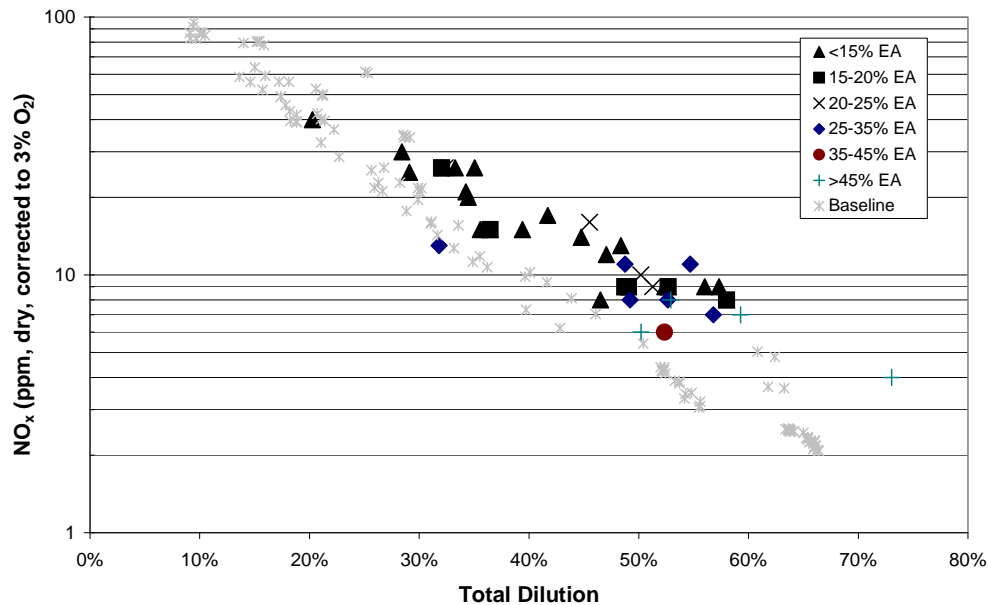
excess air and flue gas relative to the mass flow of air required for stoichiometric combustion. The excess air data from Figure 4 are presented as baseline data in Figure 5 for comparison.

FGR use in boilers is motivated by the desire to achieve low  $\text{NO}_x$  without sacrificing thermal efficiency. The experience gained while operating with high levels of flue gas proved to be beneficial in the early stages of this project. At ULNB conditions, the  $\text{O}_2$  level of the air-flue gas mixture can be on the order of 16-17%. This approaches the 15-17%  $\text{O}_2$  level considered to be typical of a gas turbine. Also, in addition to matching the burner  $\text{O}_2$  to TEG  $\text{O}_2$ , the chemical makeup of the two streams with respect to all of the major gas species (nitrogen, water vapor, carbon dioxide, and oxygen) is similar. In this important aspect of burner design, the existing boiler burner and the proposed duct burner are nearly identical.



**Figure 4. ALZETA CSB  $\text{NO}_x$  emissions plotted versus excess air over a range of burner heat release rates. Higher excess air corresponds to lower flame temperature and lower  $\text{NO}_x$ .**

Despite the similarity between current ULNB combustion with FGR and the proposed duct burner, one important difference is that the TEG is a much higher temperature stream than what is typical of a boiler air-FGR mixture. The typical TEG temperature is 950°F, while the typical ULNB air-FGR mixture temperature is on the order of 200°F. Since there is a known relationship between flame temperature and  $\text{NO}_x$ , with a higher flame temperature causing higher  $\text{NO}_x$ , this higher gas temperature prior to combustion is a concern.



**Figure 5. NO<sub>x</sub> versus total dilution for a range of excess air and FGR conditions. Dilution is calculated on a mass flow basis relative to stoichiometric air. “Baseline” is Figure 3 data.**

### ***Technical Challenges Studied in This Project***

A novel duct burner system was developed based on existing fully premixed surface stabilized burner technology that expanded on the current knowledge base by optimizing burner design for use in the duct burner application. This is a radical departure from “grid” designs that have been used for decades. Significant differences between the duct burner application and existing premixed surface burner applications present knowledge gaps that had to be addressed including:

- **Burner Performance at Very High Premix Temperatures** – Maximum premix temperatures for current generation boiler ULNBs are on the order of 200°F. ALZETA has successfully tested and operated premixed surface burners at temperatures as high as 1000°F, but not in any commercially available equipment. Additional R&D was required in the early stages of the project.
- **Burner Performance When Optimized for Low Pressure Drop Operation** – Due to the critical negative impact that system back pressure has on turbine performance, a burner designed for supplemental firing must be optimized to operate at a lower pressure drop than conventional burners. The duct burner must be designed to operate with nominally 1/2 the pressure drop of current industrial ULNBs. Reducing the pressure drop across the burner can negatively affect the following:
  - **Mixing** of fuel and air or fuel, air, and flue gas becomes more difficult, and good mixing is especially important in ULNB applications.

- **Flow uniformity** across the burner is critical to optimum performance. Flow uniformity typically involves a design trade-off between achieving the desired uniformity and minimizing pressure losses. Improved designs can improve the balance between uniformity and pressure.

### 1.2.2. Economics

The positive energy and environmental impacts of this project work are estimated in this section (see Table 4). The Innovative Energy Systems (IES) Energy Savings Tool (Reference 3) was used as the basis of this estimate, and the inputs and outputs from that program are presented below. The basis for this tool was a DOE-sponsored program with the chemical industry, and while the focus for this group was the chemical industry, the methodology is believed to be applicable across the range of industries that are of interest to researchers. The primary source for market data for CHP utilization was Reference 4.

**Table 4. Energy and environmental impacts**

User Inputs			
Sector:	chemicals	% Energy Savings Natural Gas:	14.0%
% of Market Impacted:	50.0%	% Energy Savings Electricity:	0.0%
Annual Growth Rate:	5.0%	% Energy Savings Coal/Coke:	0.0%
Year of Introduction:	2008	% Energy Savings Fuel Oil:	0.0%
Market Penetration Curve:	20 Year Market Saturation	Solid or Liquid Wastes:	0 lbs per tons of product
Technology Opportunity Area:	Crosscutting/Systems Approach	Energy Opportunity Area:	Combined heat and power systems
Energy Recovery Quality:	NA	Multiple Industry Impacts:	Chemicals; Petroleum refining; Pulp & paper; Food processing
Capital Cost Conventional System:	\$7,500,000	Capital Cost New System:	\$7,600,000
Non-energy Annual Cost Conventional System (\$):	\$750,000	Non-energy Annual Cost New System (\$):	\$750,000
Technology Lifetime in Years	20	Non-Combustion Air Pollutants:	0 lbs per ton of product

## Energy Impacts for chemicals

(Based on the input provided above, this technology will have the following impacts :)

	2005	2010	2015	2020
<b>MARKET PENETRATION *</b>	N/A	1.9%	5.4%	14.7%
<b>MARKET (MW)</b>	N/A	24.25	87.96	305.60
<b>ENERGY SAVINGS</b>				
Natural Gas Energy Savings (trillion Btu)	N/A	4.46	16.18	56.22
Electricity Savings (trillion Btu) **	N/A	0.00	0.00	0.00
Coal Energy Savings (trillion Btu)	N/A	0.00	0.00	0.00
Fuel Oil Energy Savings (trillion Btu)	N/A	0.00	0.00	0.00
<b>TOTAL ENERGY SAVINGS</b>	N/A	4.46	16.18	56.22
<b>Energy Cost Savings (million \$)</b>				
Natural Gas	N/A	18.06	77.99	274.92
Electricity	N/A	0.00	0.00	0.00
Coal	N/A	0.00	0.00	0.00
Fuel Oil	N/A	0.00	0.00	0.00
<b>TOTAL ENERGY COST SAVINGS</b>	N/A	18.06	77.99	274.92
<b>POLLUTANT REDUCTIONS (lbs)</b>				
Carbon (MMTCE/yr)	N/A	0.06691	0.24272	0.84328
Nitrogen Oxides (NOX)	N/A	1,048,314	3,802,577	13,211,377
Sulfur Oxides (SOX)	N/A	4,461	16,181	56,219
Carbon Monoxide (CO)	N/A	281,037	1,019,414	3,541,773
Volatile Organic Compounds (VOCS)	N/A	26,765	97,087	337,312
Particulates	N/A	0	0	0
Other (million lbs)	N/A	0	0	0
<b>Return on Investment</b>	0%			
<b>% Impact on Technology Opportunity Area</b>	N/A	0.70	2.55	8.87
* % of possible market impacted as shown in user inputs, not the total market. ** Includes electricity generation and transmission losses.				

## User Explanations

Technology Description:	The High Efficiency Duct Burner will be used in industrial CHP systems to boost CHP efficiency to the efficiency currently achieved by state-of-the-art boilers (80% based on HHV).
Market Percentage:	Turbine and duct burners can be added upstream of conventional boiler, so boiler does not have to be replaced.
Introduction Year:	Field demonstration is planned for late 2007. First sale will be the following year.
Energy Impacts Percentages:	Thermal efficiency will be increased for 70% current typical HHV efficiency to 80%. Electricity will be more effectively used by being generated on site, but this is not addressed.
Other Wastes and Pollutants:	None

### ***Summary of Inputs to IES Energy Tool***

Industrial CHP systems can be applied in a broad range of applications that include chemical plants, refineries, paper mills, and any large industrial site that uses both electricity and heat. The current installed base of CHP systems in the United States is approximately 50 gigawatts (GW) based on U.S. Combined Heat and Power Association estimates. This includes some very large sites and some sites that do not burn natural gas. Hence, the market for CHP was revised downward to 10 GW to provide a conservative estimate. The market growth rate was set at 5%, which is also conservative compared to some other estimates, with an estimated 20-year horizon for market saturation. Capital cost was for the complete CHP system including a 5 MW turbine, duct burners, and HRSG. Cost was estimated based on \$1,500 per kW installed for the system. The installed cost of the turbine alone would be on the order of \$1,000 per kW. The \$1,500 amount includes the remaining system components.

### ***Reduced Energy Usage for Equivalent Productivity***

For a given facility, the estimated fuel usage can be reduced by 11% relative to the current typical CHP installation. This was based on an estimated HHV thermal efficiency of 80% versus 72% for a typical current system. When using the table, the comparison was for the improved CHP system versus a current typical system. This resulted in energy savings of 56 trillion Btu annually in 2020. Another ALZETA assumption, not quantified in the IES spreadsheet, is that improved duct burners would result in more widespread use of CHP, which will save energy by replacing low-efficiency electric power generation and transmission with higher efficiency on-site generation.

### ***Environmental Benefits***

The environmental benefits are very significant, with annual NO<sub>x</sub> emissions reduction by 2020 being 13 million tons per year, CO reduction being 3.5 million tons per year. Carbon

emissions are also reduced due to the more efficient conversion of fuel to heat and power, resulting in less combustion of fossil fuel.

### ***Public and Private Benefits of Reduced Peak Load on the Grid***

A more difficult impact to quantify is the private benefits that accrue as more large industrial users generate a larger fraction of their own power. This will either eliminate the need for additional central power plants or will reduce the demand for power to be transmitted to end users. The net result will be lower demand on the existing power infrastructure. The greatest problems on the grid can be expected to occur at times of peak demand. The addition of cogeneration at industrial facilities can significantly reduce peak demand, which benefits not only the industrial user; it also benefits other customers on the grid by relieving the required transmission capacity. This is a very real but difficult-to-quantify benefit of industrial CHP systems.

### ***Economic Viability of the Proposed Product***

The duct burner technology developed under this project should have a capital cost that is estimated to be \$100,000 higher than that of a current duct burner system but with no incremental operating cost increase. However, the proposed high-efficiency system should save the end user \$500,000 per year in fuel, based on 150 MMBtu/hr heat input, natural gas as fuel at \$8/MMBtu, and 50% utilization factor (using the system for half of its rated capacity). Consequently payback for this system is estimated to be less than one year. (Note that the IES Tool did not calculate this. Possible problems were the large cost of the total system compared to the “incremental” cost of the duct burners; or the fact that the total “system” allowed the end user to not purchase 5 MW of power. The power savings was not accounted for in the authors’ comparison of two CHP systems.) Whatever the reason, the proposed product is economically very viable with a payback of less than one year relative to the standard duct burner.

## **1.3. Project Overview**

This project was divided into three development phases that progressed from the laboratory to testing under actual field conditions. During each phase, specific performance characteristics were analyzed to establish design requirements for the subsequent phase. This minimized the risks at each stage of development and helped ensure that each phase’s development activities represented significant steps toward a commercial product. Work was performed over a 4 ½-year period, starting June 2004. Prototype and pilot-scale activities were performed at ALZETA Corporation in Santa Clara, California. Full-scale field tests took place at Cenveo Anderson Lithograph in City of Commerce, California, and Stelter & Brinck, Harrison, Ohio.

### ***1.3.1. Project Phases***

Project phases included:

- Prototype phase, in which burner materials were evaluated and key parameters were established for burners operating downstream of a turbine-generator.

- Pilot phase, which focused on the design of an industrial-sized module for the ultra-low NO<sub>x</sub> supplemental firing burner and proving its operation.
- Full-scale field test phase, where multiple burner modules were tested under industrial conditions.

The prototype phase was especially important in that if burner operating parameters could not be determined for the high-temperature, low-O<sub>2</sub> conditions that represent a typical TEG environment, it would be difficult to continue. However, if it could be demonstrated that the materials and general operating concept of the prototype supplemental firing burner could simultaneously achieve ultra-low NO<sub>x</sub> and CO emissions, uniform and stable combustion, and low pressure drop when operating under TEG conditions, continuing to develop a design for industrial scale would be highly justified.

Likewise, demonstrating that performance goals could be achieved when key operating parameters were applied to an industrial-scale module design was essential to successfully completing the pilot-scale phase. Positive results at this scale would justify building a matrix of modules and proceeding to the field testing phase.

### **1.3.2. Problems Encountered**

Problems occurred during the field test phase when it became apparent that real TEG conditions and the CHP environment were harsher and much more variable than anticipated. This led to cyclic loading and vibrations that caused structural failure. A redesign and second field test were therefore necessary to prove the technology.

### **1.3.3. Project Accomplishments**

Overall the project demonstrated that ultra-low NO<sub>x</sub> can be achieved in the low-O<sub>2</sub>, high-temperature environment typical of turbine exhaust gas. The NO<sub>x</sub> emission levels that can be achieved will meet BACT requirements in the most stringent air districts in the United States. In addition, it was shown that a strong correlation exists between NO<sub>x</sub> emissions and adiabatic flame temperature, making it possible to predict emissions and set operating conditions as TEG conditions vary. And finally, a rugged and modular design was developed to withstand the varying flow field and vibrations of TEG applications.

## **1.4. Report Organization**

This report presents the key findings of each phase. Section 1 presents the Executive Summary, and Section 2 provides the Introduction and Project Background, Goals, and overall project organization. The results of the prototype and pilot scale phases are presented in Section 3. Section 4 highlights the results of field testing phase. And finally, Section 5 presents the Conclusion and Recommendations for the project, including several market opportunities.

## **2.0 Project Approach**

The project was divided into three phases as described below.

### **2.1. Prototype Phase**

The prototype phase tested burner materials at laboratory scale to characterize emissions and combustion performance on simulated turbine exhaust gas (TEG). If successful and baseline operating parameters could be established, the project would proceed to the next phase: pilot scale.

#### **2.1.1. Objective**

The objective of the prototype phase was to demonstrate that an ultra-low NO<sub>x</sub> supplemental firing burner based on ALZETA's CSB ultra-low NO<sub>x</sub> burners could operate predictably on simulated TEG. The Phase investigated key performance parameters such as NO<sub>x</sub> and CO emissions, pressure drop across the burner surface, and evaluated burner geometries compatible with typical duct cross-sections and burner surface flux constraints.

#### **2.1.2. Geometry and Scale**

Tests were performed with cylindrical, conical, and wedge-shaped burner elements at a nominal firing rate of 100-200 MBtu/hr. At this scale tests can be completed in the laboratory in a rig that provided excellent visual access to the burner. It also provided a suitable environment for demonstrating burner flame stability and emissions over a range of operating conditions that are representative of the turbine exhaust gas (TEG) environment. Prior experience has shown that prototype testing with materials based on ALZETA's CSB ultra-low NO<sub>x</sub> burner are readily scaled-up to larger systems. Similar scale-up results were expected for TEG environments.

The key performance parameters to be investigated were:

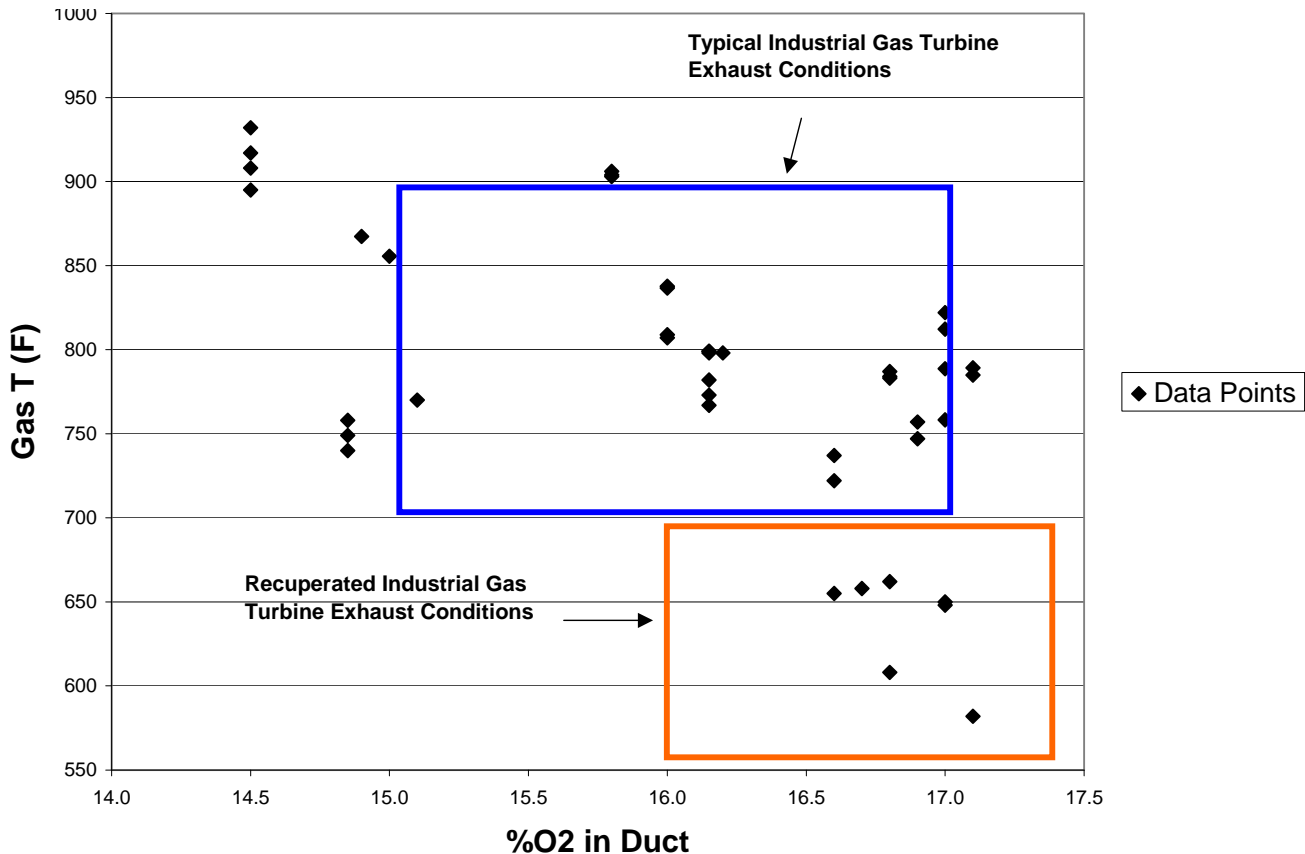
- Ultra-low emissions of NO<sub>x</sub> and CO.
- Low-pressure drop across the burner.
- Burner geometry compatible with the duct cross-section and burner surface flux constraints of this application.

At the prototype firing rate selected, the tests were approximately 1/40 of a full-scale burner module, although the specific capacity of a full-scale module was not defined until the completion of the pilot scale phase.

#### **2.1.3. Operating Conditions**

Test data were to be recorded at points that represented typical turbine outlet conditions. The initial test matrix was presented in Deliverable 6 and proposed varying the TEG gas temperature between 700°F and 900°F and the oxygen concentration between 15% and 17% dry O<sub>2</sub>. Based on interest from a recuperated engine manufacturer, the test matrix was expanded to lower temperature and slightly higher TEG O<sub>2</sub> conditions to test at conditions more representative of recuperated engine exhaust. The targeted range of TEG temperature and O<sub>2</sub> conditions is presented as Figure 6.

At specific TEG temperature and O<sub>2</sub> conditions (the inlet conditions to the supplemental burner), it was proposed in Deliverable 6 to operate the supplemental burner at exhaust conditions of 4-6% O<sub>2</sub>. As a result of promising early test results, this operating range was extended to include points between 2% and 6% O<sub>2</sub>. During testing it was also determined that controlling the test rig to all combinations of temperature and O<sub>2</sub> was difficult to achieve. However, tests were conducted over a broad operating range, including some points outside of the proposed test windows, and at all conditions that are of interest in this project.



**Figure 6. Test matrix and test conditions for prototype tests.**

#### **2.1.4. Results**

##### ***Emissions***

NO<sub>x</sub> emissions have been previously shown to vary as a function of excess combustion air and any other diluents in the premix stream, combustion air preheat, and other variables. It has also been demonstrated that NO<sub>x</sub> emissions do correlate well to a single parameter, adiabatic flame temperature (AFT), which is a single variable that incorporates most of the other input variables. Therefore, for every test point, the AFT was calculated and used as the dependent variable in all emissions plots. The AFT calculations were performed using ALZETA's in-house equilibrium code, CHEM. CHEM requires gas composition, excess air, and temperature as input values. The input gas concentration and excess air are both calculated utilizing the measured percentage O<sub>2</sub> in the gas stream and assuming combustion of methane since the

exact composition of the natural gas used in testing was unknown. The temperature was measured with a thermocouple. CHEM uses these input values and Joint Army Navy Air Force (JANAF) thermochemical properties to calculate the heat of reaction and equilibrium concentrations of 30 gas species that represent by-products of the combustion reaction.

The test results confirmed that for the prototype supplemental firing burner, NO<sub>x</sub> emissions do correlate with the adiabatic flame temperature. Figure 7 shows this relationship and that the goals of reaching NO<sub>x</sub> emissions below 9 ppm at 3% O<sub>2</sub> (3 ppm at 15% O<sub>2</sub>) were achieved. Keeping NO<sub>x</sub> emissions below 9 ppm requires an AFT below 2700°F. Flame temperatures below 3050°F resulted in NO<sub>x</sub> emissions of 27 ppm at 3% O<sub>2</sub> (9 ppm at 15% O<sub>2</sub>), which is considered to be excellent low-NO<sub>x</sub> performance but does not meet the performance goals of this project. Test points meeting this emissions level include all test points with 4 to 6% O<sub>2</sub> in the stack, and test points with 2% O<sub>2</sub> in the stack and 15% O<sub>2</sub> in the duct. Flame temperatures below 2700°F include most test points with 6% O<sub>2</sub> in the stack and test points with 4% O<sub>2</sub> stack and 15% O<sub>2</sub> in the duct.

For comparison, the test results are matched-up to earlier gas turbine injector results that were used to estimate supplemental burner performance as shown in Figure 8. These data were taken with preheated air at temperatures between 350 and 500°F and pressures between two and eight atmospheres. These conditions provided similar AFTs, and therefore the emissions results are comparable. Although the supplemental burner tests show higher NO<sub>x</sub> emissions at a specific AFT, the differences are relatively small. Of potentially greater interest the supplemental burner had a much broader operating range at the low temperature end of the range, which provides the potential for achieving lower emissions levels with stable burner operation. In addition, there was no evidence that CO emissions would be a problem anywhere in the authors' proposed range of operation.

## Supplemental Burner Emissions

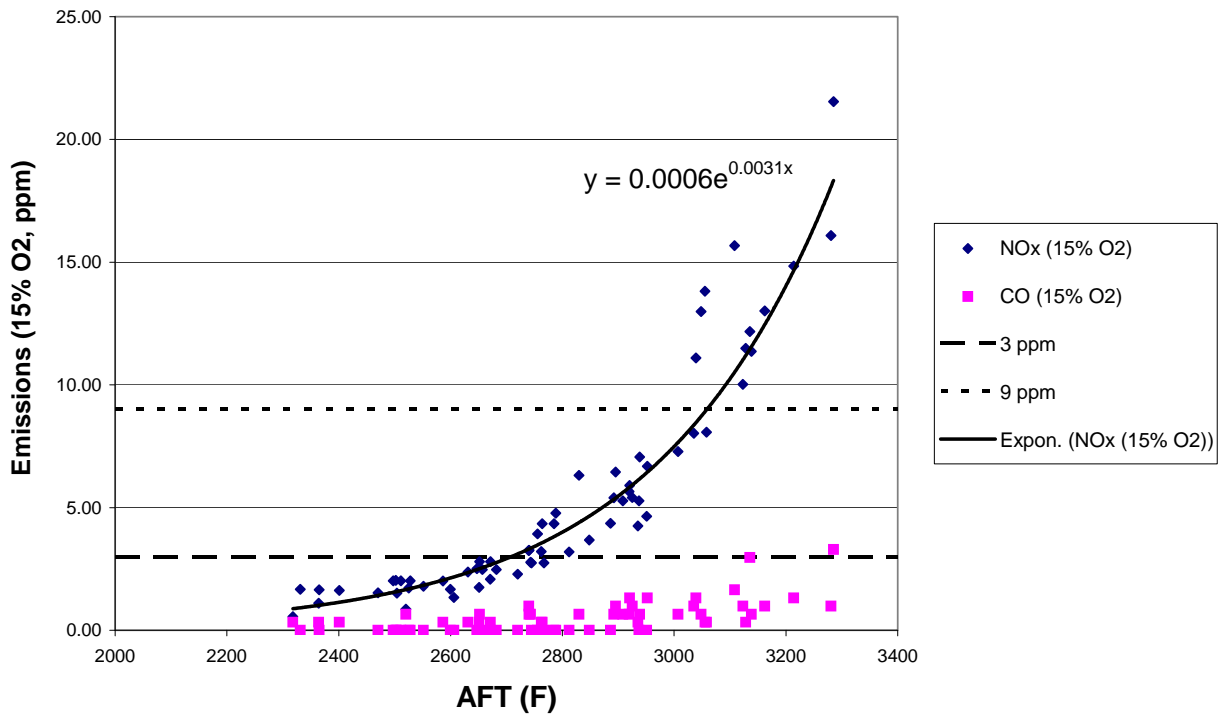


Figure 7. Effect of adiabatic flame temperature (AFT) on NO<sub>x</sub> emissions.

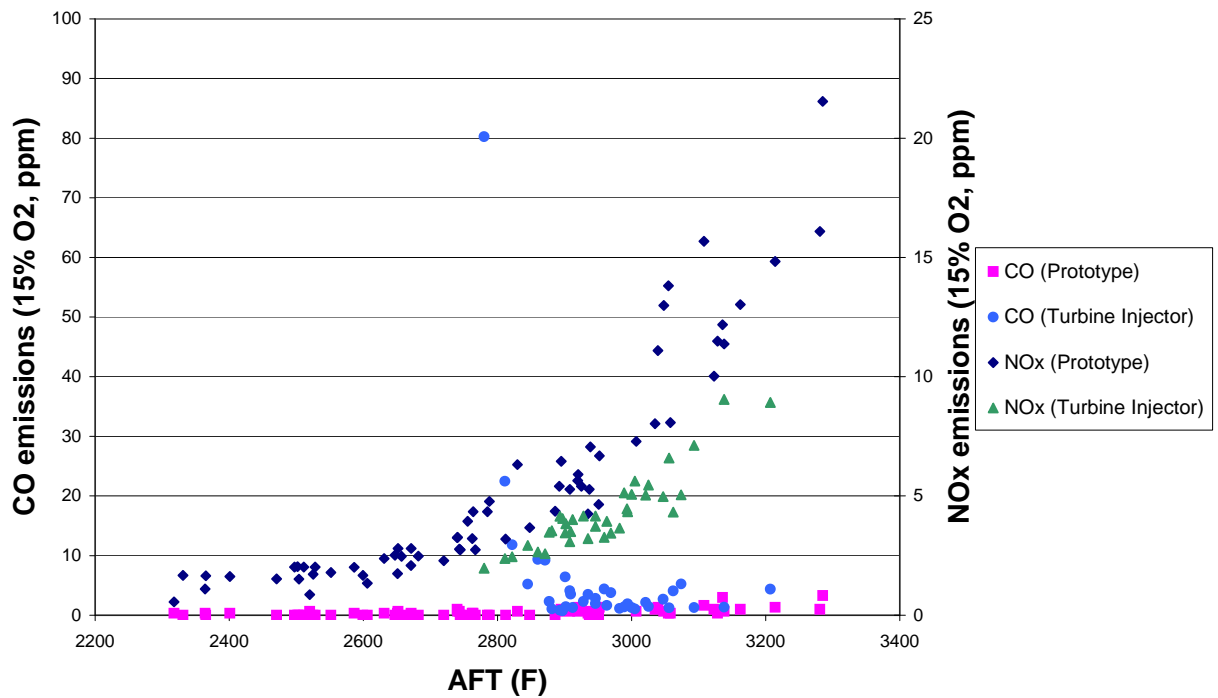


Figure 8. Effect of adiabatic flame temperature (AFT) on CO and NO<sub>x</sub> emissions.

### **Pressure Drop**

After achieving emissions requirements, the next most critical performance parameter was to minimize pressure drop. The pressure drop was shown to correlate with the dynamic pressure, which is defined as  $\frac{1}{2}$  the density multiplied by the velocity squared.

$$\text{Dynamic Pressure} = (\rho \cdot v^2) / 2g_c$$

Other tests showed that pressure drop through the surface burner scales linearly with dynamic pressure over the operating range of interest, and the goal of keeping the pressure drop below 3 in. w.c. was achieved over all conditions by keeping the dynamic pressure below 0.052 in. w.c.

### **Geometric Compatibility**

To keep the technology consistent with standard industrial practice, the general design approach for the ultra-low NO<sub>x</sub> supplemental firing burner was to use multiple burner modules, which could be mounted in a grid-like arrangement within the ductwork between the turbine exit and the HRSG inlet. Based on existing information on HRSG inlet dimensions, and the authors' prototype test results, which demonstrated a maximum burner surface flux that could be achieved without loss of flame stability, a burner surface-to-cross sectional area ratio of approximately 1.3:1 was calculated. There were a number of ways to configure a burner that has more surface area than cross sectional area.

Two burner geometries were analyzed to evaluate the ability to scale to pilot and ultimately to full scale. The first is a wedge design that included a half-angle of 30°. This design provides a burner surface area that is two times the burner cross-sectional area.

A second design was a truncated cone. With a two-foot diameter base and a six-inch diameter at the end, the truncated cone has a surface area to cross-sectional area ratio greater than 2, specifically a surface area of 8.388 ft<sup>2</sup> when installed in a 2' x 2' duct. However, to avoid flame interaction, the cones would need to be placed in rows with six-inch spaces on all sides. This spacing will yield a final surface area to cross-sectional area ratio of 8.4:6.25 or 1.34:1.

To verify that the wedge and cone geometry would have the same emissions and pressure drop performance, the two concepts were compared, knowing that if the two burner geometries have the same velocity profile, then they will have the same pressure drop. An arbitrary flow of 100 ft<sup>3</sup>/min was selected for both cases. For every 10% increment of travel down the length of the burner, the amount of surface area that has been passed was calculated. Then the amount of remaining flow was calculated by assuming a constant flux through the burner surface. The velocity was then calculated by dividing the remaining flow by the interior area of the burner. The conclusion was that the velocity is constant within either geometry, and the two should be interchangeable.

Then to check that the pressure drop will remain the same after scaling-up to full-size, calculations were performed with the exit conditions from three industrial gas turbines. The three turbines chosen were the Kawasaki GPB-15X, Solar Turbines Taurus 70, and General Electric Frame 7F. Table 5 shows what typical exit conditions, duct sizes, and velocities would

be for these three turbines if a series of duct burner modules were installed. The table shows that all of the scaled-up conditions are very similar to the prototype test conditions. Because the conditions are similar, the dynamic pressure and therefore pressure drop is expected to fall into the same range and stay below the allowable 3 in. w.c. after scale-up.

**Table 5. Typical gas turbine / HRSG conditions relative to prototype burner test conditions**

<b>Duct Burner System</b>	<b>Property</b>	<b>Wedge</b>	<b>Truncated Cone</b>
<b>Prototype Rig</b>	<b>TEG Mass Flow (lb/s)</b>	0.129	0.129
	<b>Temperature (F)</b>	800	800
	<b>Duct Dimensions</b>	5" x 5"	5" x 5"
	<b>Cross Sectional Area (sq ft)</b>	0.17	0.17
	<b>Duct Velocity (ft/s)</b>	24.0	24.0
	<b>Burner Surface Area (sq ft)</b>	0.20	0.20
	<b>Face Velocity (ft/s)</b>	20.8	20.8
<b>Kawasaki GPB-15X</b>	<b>TEG Mass Flow (lb/s)</b>	17.6	17.6
	<b>Temperature (F)</b>	995	995
	<b>Duct Dimensions</b>	6' x 5'	5.5' x 5.5'
	<b>Cross Sectional Area (sq ft)</b>	30.0	30.25
	<b>Duct Velocity (ft/s)</b>	21.9	21.7
	<b>Burner Surface Area (sq ft)</b>	40.0	33.6
	<b>Face Velocity (ft/s)</b>	16.4	19.6
<b>Solar Taurus 70</b>	<b>TEG Mass Flow (lb/s)</b>	59.0	59.0
	<b>Temperature (F)</b>	913	913
	<b>Duct Dimensions</b>	10' x 10'	10.5' x 10.5'
	<b>Cross Sectional Area (sq ft)</b>	100 ft <sup>2</sup>	110.25 ft <sup>2</sup>
	<b>Duct Velocity (ft/s)</b>	20.8	18.8
	<b>Burner Surface Area (sq ft)</b>	120	134
	<b>Face Velocity (ft/s)</b>	17.3	15.4
<b>GE Frame 7F</b>	<b>TEG Mass Flow (lb/s)</b>	991	991
	<b>Temperature (F)</b>	1249	1249
	<b>Duct Dimensions</b>	40' x 40'	40.5' x 40.5'
	<b>Cross Sectional Area (sq ft)</b>	1600	1640
	<b>Duct Velocity (ft/s)</b>	27.1	26.5
	<b>Burner Surface Area (sq ft)</b>	2080	2147
	<b>Face Velocity (ft/s)</b>	20.9	20.2

## Conclusions

Test results provided two correlations that can be used to model the NO<sub>x</sub> emissions and pressure drop within the duct burner system. The NO<sub>x</sub> emissions are shown to be accurately modeled as a function of AFT by:

$$\text{NO}_x(15\% \text{ O}_2)[\text{ppm}] = 0.0006 * e^{(0.0031 * \text{AFT} [^\circ\text{F}])}$$

The pressure drop can be modeled as a function of dynamic pressure by:

$$\Delta p[\text{in. wc}] = 58.129 * (\text{Dynamic Pressure}[\text{in. wc}])$$

Both of these parameters can be scaled to larger burners and systems. In order to keep the authors' goals of sub-9 ppm NO<sub>x</sub> at 3% O<sub>2</sub> (dry) and sub-3 in. w.c. pressure drop, the flame temperature had to be kept below 2700°F, and the dynamic pressure had to be kept below 0.052 in. w.c. in pilot and full-scale designs.

Two geometries for scale-up were considered, a wedge and a cone. In order to verify that emissions and pressure drop performance would remain the same, calculations were performed. It was determined that emissions performance would be unaffected because it is controlled by the AFT, which is independent of geometry. The pressure drop performance will also remain the same because the velocity is constant inside either geometry.

Scale-up calculations of a duct burner with two potential burner geometries were performed under typical exhaust conditions and duct sizes of three industrial gas turbines. These calculations demonstrated that the prototype test rig conditions closely approximated those of a typical industrial gas turbine exhaust duct. The temperatures and gas velocities in the industrial turbines and the test rig were all within a close range. The prototype inlet temperature was 700-900°F, and the industrial turbines' temperatures were between 914°F and 995°F. Prototype inlet velocity was 20.8 ft/s, and the industrial turbine velocities were between 15.4 and 20.9 ft/s. Consequently, the same correlations for emissions and pressure drop were expected to hold under scaled-up conditions.

## 2.2. Pilot Phase

Prototype testing defined modular design geometries for the ultra-low NO<sub>x</sub> supplemental firing burner. In this phase, a larger scale design was tested and optimized under simulated TEG conditions to demonstrate that the design could meet full-scale performance goals. If successful, multiple modules would be fabricated and tested at full scale under real TEG conditions.

### 2.2.1. Objective

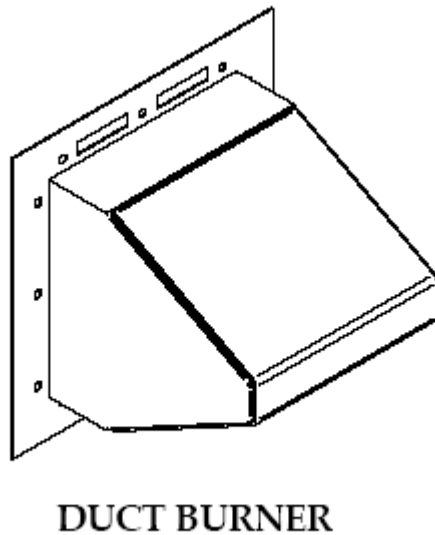
Pilot scale testing had the following objectives:

- Demonstrate stable burner performance when operating in a simulated gas turbine exhaust stream with nominal conditions of 15% stack O<sub>2</sub> and 800°F temperature.

- Demonstrate this stable operation with NO<sub>x</sub> emissions of 3 ppm or lower when corrected to 15% O<sub>2</sub> dry, and CO emissions of fewer than 16.5 ppm corrected to 15% O<sub>2</sub> dry.
- Demonstrate the stable operation and low emissions with a burner pressure drop of 3 in. w.c. for the complete burner system

### **2.2.2. Geometry and Scale of Burner**

Since typical duct burner systems are made up of multiple modules, the pilot scale burner was designed to be equivalent to a single module based on a version of the prototype geometries. Testing was done with a wedge-shaped configuration instead of a conical burner as the wedge shape yields a higher burner surface area to cross-section area when flame interaction between burners is considered. The test burner configuration is shown in Figure 9 below:



**Figure 9. Duct Burner**

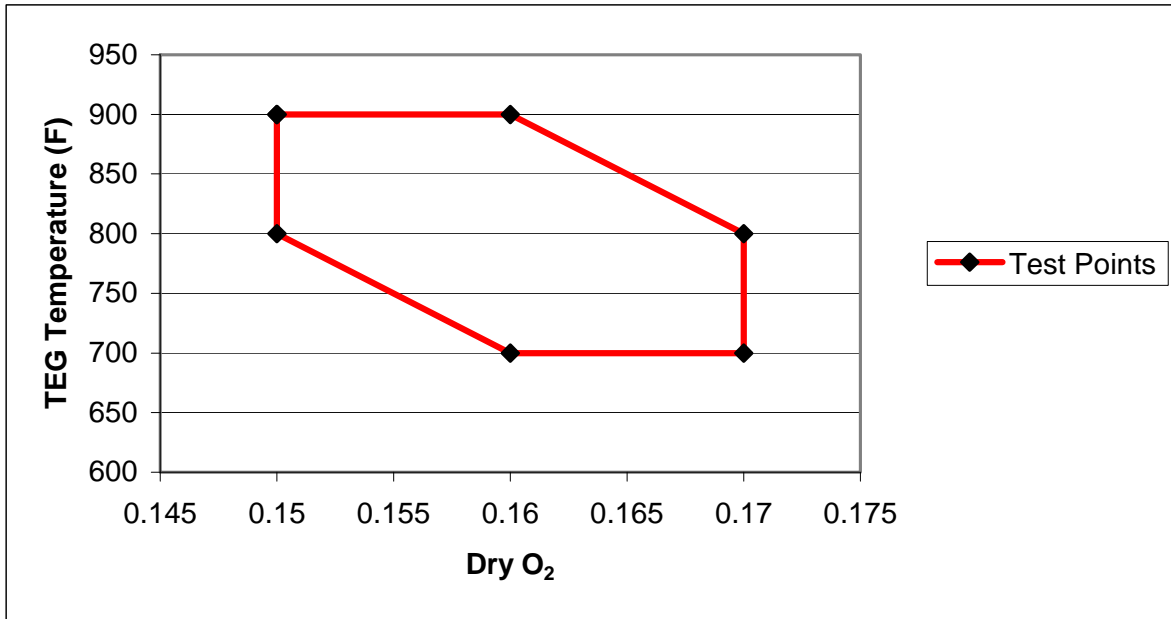
### **2.2.3. Operating Conditions**

In these tests ALZETA used a gas turbine simulation rig to create typical TEG conditions upstream of the burner. Testing involved the following:

- Burner inlet temperature was varied between the extremes of 700°F to 900°F, in increments of 100°F.
- Burner inlet oxygen content was varied between 15% and 17% O<sub>2</sub>, in increments of 1% O<sub>2</sub>.
- Stack O<sub>2</sub> content was varied between 2% and 6% dry O<sub>2</sub>, in increments of 2%.
- Burner inlet flow rate was varied between 0.75 and 1.25 times the design value of 925 scfm.

The conditions fit the test matrix shown in Figure 10. For every inlet condition shown, data were intended to be taken at 2%, 4%, and 6% O<sub>2</sub> in the stack.

The gas turbine simulation rig consists of two burners fired in series. The first burner serves as the gas turbine combustor simulator, providing burner exhaust at conditions representative of a TEG stream. An example of these “simulated turbine exhaust” conditions would be a hot vitiated air stream with a temperature of nominally 800°F and 15 percent oxygen content. The fired duty of the burner generating the simulated turbine exhaust ranged from 400 to 700 MBtu/hr, depending on operating conditions. The exhaust conditions from this burner define the inlet conditions for the ultra-low NO<sub>x</sub> supplemental firing burner pilot module or duct burner.



**Figure 10. Test matrix.**

Prior to entering the duct burner, natural gas is injected into this stream, mixed, and combusted by the duct burner at nominally 2 MMBtu/hr. The exhaust from the duct burner then flows into a small heat recovery steam generator where some gas is recycled back to the system to optimize test conditions, and the remainder leaves the system.

The key performance parameters investigated were:

- Burner stability when operating in a simulated gas turbine exhaust stream with nominal conditions of 15% O<sub>2</sub> and 800°F.
- Stable operation with NO<sub>x</sub> emissions of 9 ppm or lower when corrected to 3% O<sub>2</sub> dry, and CO emissions of fewer than 50 ppm corrected to 3% O<sub>2</sub> dry.
- Stable operation and low emissions with a burner pressure drop of 3 in. w.c. for the complete burner system.

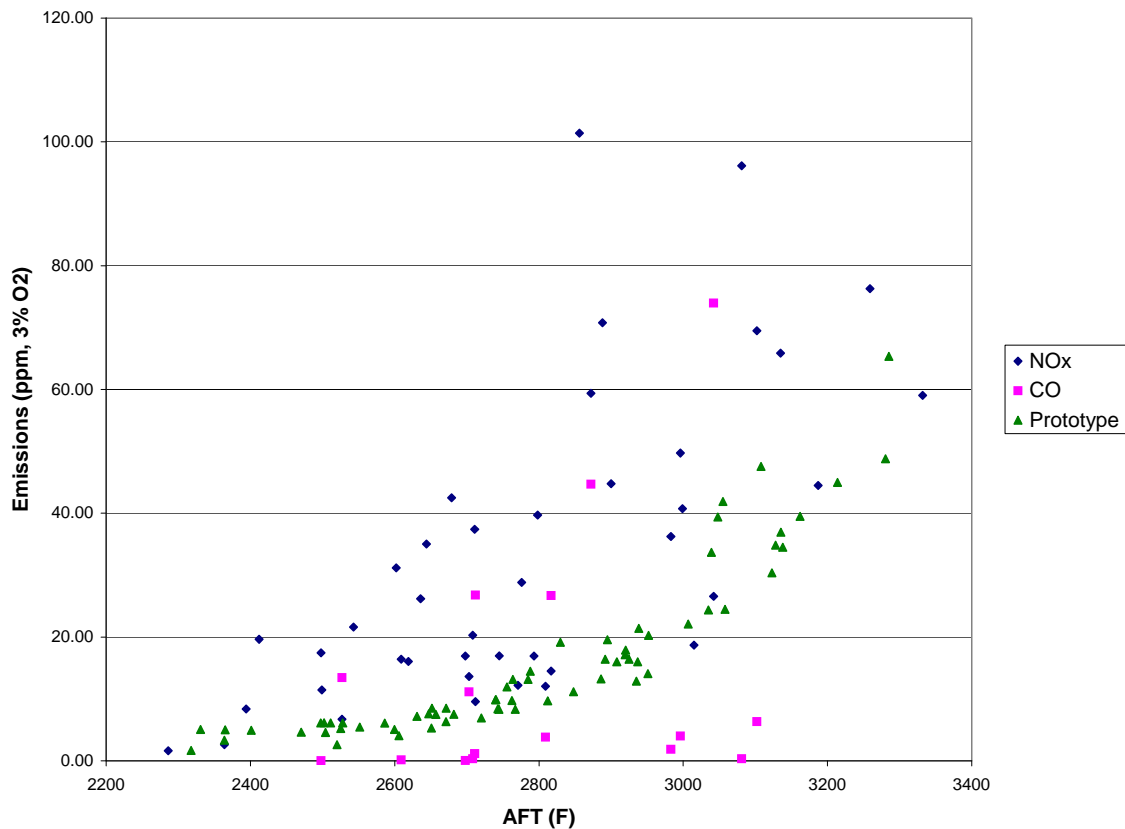
#### **2.2.4. Results**

Previously it was shown that NO<sub>x</sub> emissions correlate well to a single parameter, adiabatic flame temperature (AFT). Therefore, for every test point the AFT was calculated and used as

the dependent variable in all emissions plots. The AFT calculations were performed using ALZETA's in-house equilibrium code, CHEM. The procedure for calculating AFT was described in Section 2.1.4.

### **Emissions**

During an initial test sequence, the emissions data were not consistent with the earlier prototype test data, such that NO<sub>x</sub> and CO emissions were much higher than what was expected based on prototype test results. In addition the pilot scale data had significantly more scatter when plotted versus AFT. Data were taken across the entire test matrix and are presented in Figure 11. It was also observed during this testing that the duct burner consistently appeared to be running hot and eventually was damaged.



**Figure 11. Initial effect of adiabatic flame temperature (AFT) on emissions.**

Due to these inconsistencies with the prototype test results, the burner and test rig were inspected. The cause of the unexpected performance was determined to be bypass air from the simulated turbine exhaust stream that was not premixing with the duct burner fuel. Instead, it was passing through the boiler and biasing the stack oxygen reading upward.

As part of the initial pilot scale burner design, some of the simulated turbine exhaust gas was to bypass along the top and bottom of the duct burner housing to cool the metal walls during

operation. This flow intentionally would not mix with the fuel and, therefore, would not act to dilute and cool the premix flame. The bypass air was initially assumed to be a very small portion of the flow. But after review it was determined to be a much larger fraction of the flow and therefore had a measurable impact on O<sub>2</sub> as measured in the stack, making the NO<sub>x</sub> emissions higher than they should have been. These artificially high readings lead the research team to run the burner too rich, which damaged the burner and yielded incorrect AFT calculations.

Based on these unsatisfactory results, a new burner was built and the housing modified to reduce the bypass flow. In addition, the fuel-air ratio in the duct burner was determined based on a premix measurement instead of the stack measurement. This burner was tested across the entire test matrix with both stack and premix O<sub>2</sub> measurements taken. The premix O<sub>2</sub> measurement provides an accurate way to tell the true fuel-air ratio of the burner and to calculate the AFT without bypass air interference. These tests showed that when the premix measurement was used to calculate the AFT, the emissions data were consistent with what was expected based on the prototype tests. The calculations using the stack measurement were all shifted toward lower AFTs. The NO<sub>x</sub> emissions based on the premix reading followed the exponential equation:

$$\text{NO}_x = 0.0013 * e^{0.0033 * \text{AFT}}$$

To meet the goal of 9 ppm NO<sub>x</sub> at 3% O<sub>2</sub>, the AFT for the pilot scale needed to be kept below 2650°F. This is similar to the 2700°F limit observed during the prototype tests. For all test points the CO emissions were less than 1 ppm, which is the detection limit of the analyzer and therefore are not shown on this graph. Figure 12 shows the data for these tests.

Data were taken across the entire test matrix with the new burner. The test matrixes with all inlet condition test points are shown in Figure 13. For each inlet condition, data were taken for 2%, 4%, and 6% O<sub>2</sub> in the stack.

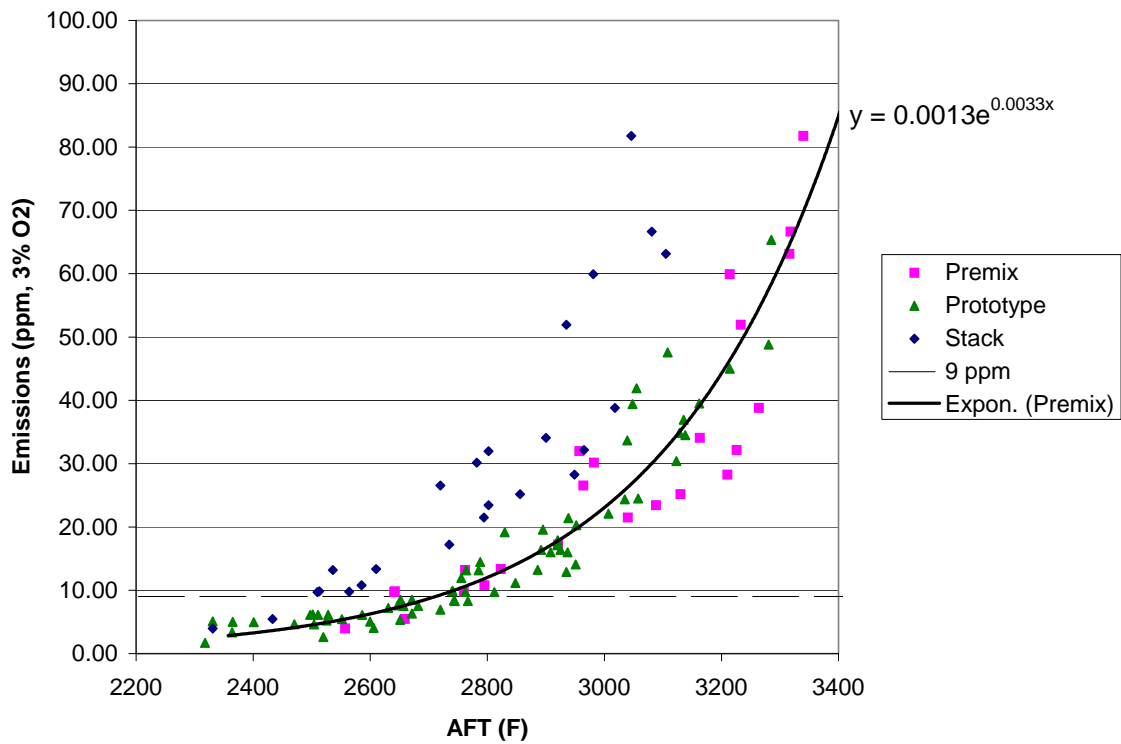


Figure 12. Accurate effect of adiabatic flame temperature (AFT) on emissions.

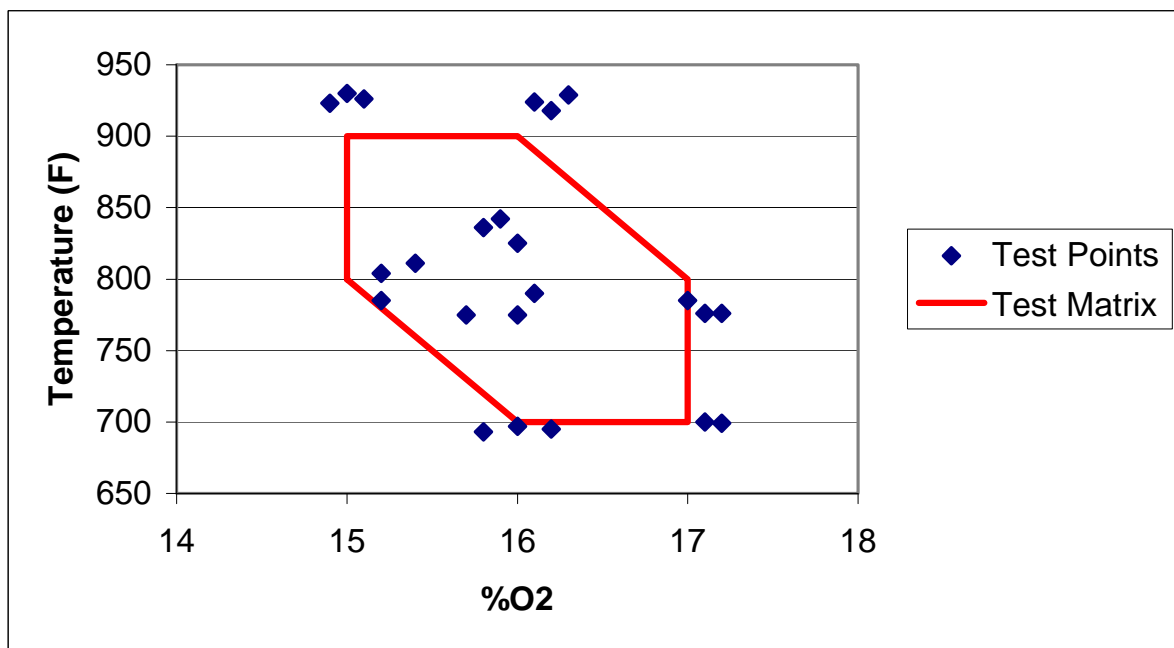


Figure 13. Test points for emissions data.

## **Pressure Drop**

After meeting emissions requirements, the next most critical performance parameter was that pressure drop across the burner be minimized. The pressure drop has been shown to correlate with the dynamic pressure, which is defined as:

$$\text{Dynamic Pressure} = (\rho \cdot v^2) / 2g_c$$

In order to calculate the dynamic pressure, the density was calculated using the ideal gas law at the measured temperature and pressure. The velocity was calculated from the known volumetric flowrate and burner surface area. The volumetric flowrate was measured in standard cubic feet per minute (scfm) and converted to actual cubic feet per minute (acfm) by taking into consideration the inlet temperature and pressure drop. The velocity was obtained by dividing the actual flowrate by the burner surface area, yielding a burner “face” velocity.

Figure 14 shows pressure drop data across the mixer and burner for both burners tested. The pressure drop data for these two burners was different due to differences in their construction. The first burner (Burner 1) consisted of a 66% open area backing plate with burner knit material wrapped over the top of it. The backing plate used with Burner 2 was only 8% open area, and therefore the pressure drop was much higher. Future plans call for the burner to have a backing plate with 26% open area, so the pressure drop should fall between the two sets of data. Because the 8% open area backing plate was able to achieve a pressure drop below the target of 3 in. w.c., there was high confidence that the 26% open area backing plate would also reach the target.

## **Conclusion**

The test results provided a correlation to model the NO<sub>x</sub> emissions within a duct burner system. The pilot scale NO<sub>x</sub> emissions are modeled as a function of AFT by:

$$\text{NO}_x [\text{ppm}, 3\% \text{ O}_2] = 0.0013 \cdot e^{0.0033 \cdot \text{AFT}}$$

To meet the goal of sub-9 ppm NO<sub>x</sub> at 3% O<sub>2</sub> dry, the flame temperature must be kept below 2650°F. The goal of keeping the CO below 50 ppm at 3% O<sub>2</sub> dry was not an issue as CO emissions was zero across all test points.

Pressure drop was shown in the past to directly relate to dynamic pressure. The pilot-scale tests had less accurate measurements on the air flows and therefore this correlation was not as obvious with the new data. Pressure drop data were taken with two different burners. One burner had 66% open area surface, and another had 8% open area surface. Although the 8% open area burner pressure drops were much higher, both burners were able to achieve pressure drops below the target of 3 in. w.c.

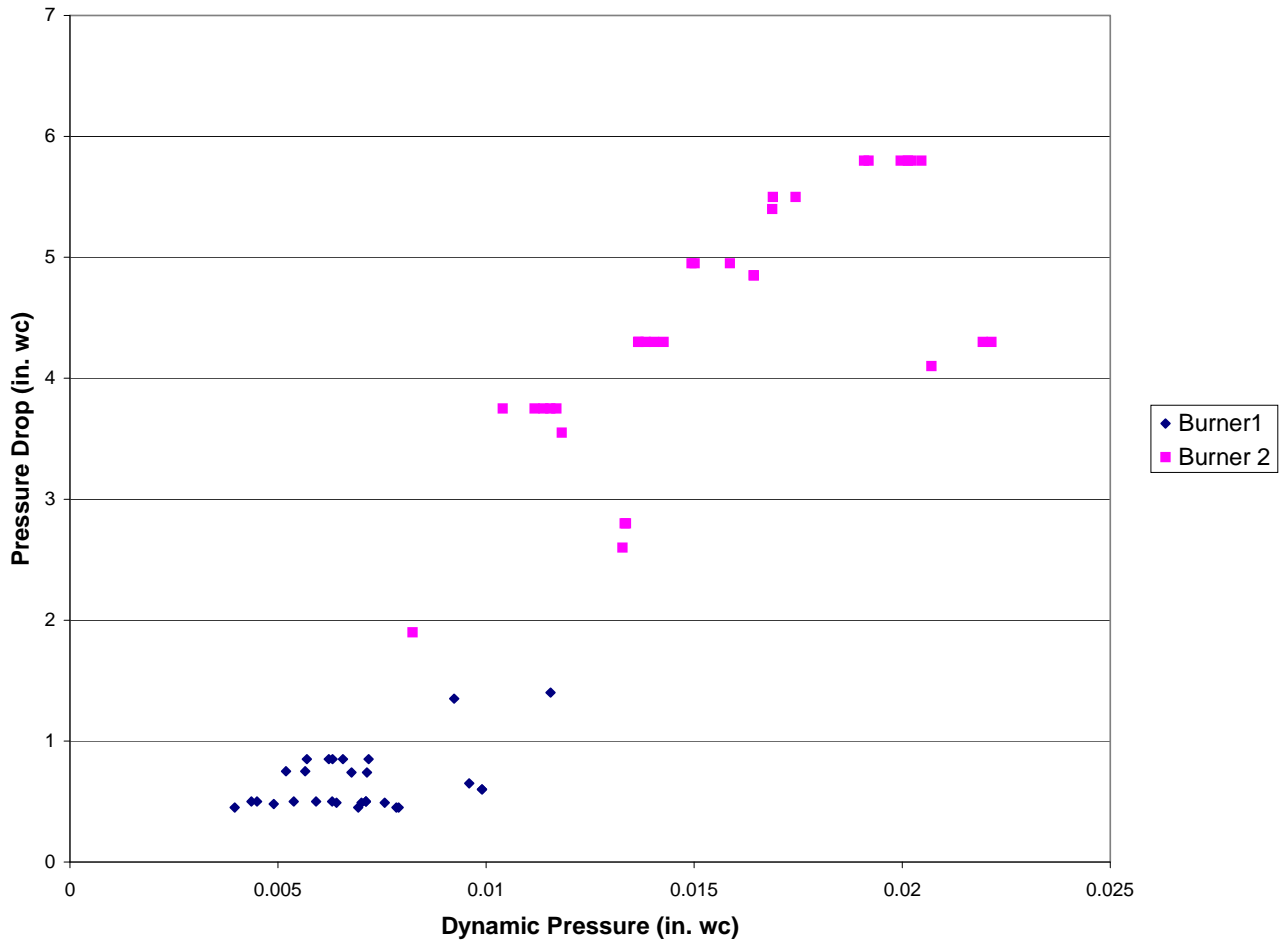


Figure 14. Effect of dynamic pressure on pressure drop.

## 2.3. Full-Scale Field Test

Based on the results of the pilot tests, a module design could be established that would allow the design of a full scale ultra-low NO<sub>x</sub> supplemental firing burner system. This phase prepared a design for a specific application with the goal of testing the system at that site.

### 2.3.1. Objective

The objective of this phase is to develop and fabricate a supplemental firing system for the field demonstration site. The new supplemental firing burner would increase overall plant thermal efficiency, improve operational flexibility, and increase the reliability of steam and electricity supplies, while reducing overall system operating costs.

### 2.3.2. Geometry and Scale of Burner System

The field demonstration hardware consisted of an array of burner modules essentially identical to the single modules tested during the pilot scale phase. Specifically, the orientation and number of these modules was defined by the physical and operating conditions at the Cenveo field demonstration site. Cenveo (See section 4.1.2.) has a 3.75 MW Rolls Royce 501

gas turbine exhausting into a duct with internal dimensions of 4.5 ft by 7 ft. When the turbine is operating, supplemental firing burners need to add 12 MMBTU/hr of additional heat to the exhaust stream in order for the HRSG and steam turbine to operate at full capacity. It was calculated that 10 modules would be required to meet Cenveo's requirements. With 10 modules, the Cenveo operating conditions for each module were similar to the ALZETA pilot-scale test conditions.

### **2.3.3. Operating Conditions**

The Rolls Royce 501 gas turbine exhaust flow is 34.0 lb/s at 1050°F. Hence, velocity through the duct where the ultra-low NO<sub>x</sub> supplemental firing burner system would be installed is 21 ft/sec—essentially the same as during the pilot-scale tests. Temperature, however, is about 250°F hotter. This would mean slightly lower dynamic pressure and, hence, somewhat lower burner pressure drop. Otherwise the environment was similar to the conditions used during the pilot tests.

## **2.4. Results**

Results are presented in the following section, Project Outcomes.



## 3.0 Project Outcomes

This section summarizes the final outcomes of the project, in which the ultra-low NO<sub>x</sub> supplemental firing burner system was tested in the field. The site selection, system design, pre-installation testing, installation and start-up, test results, and rescoping for a final test are discussed in this section.

### 3.1. Identification of Field Test Site

An important objective of this project was to demonstrate the performance of the high efficiency ultra-low NO<sub>x</sub> supplemental firing burner at a working field site for a period of six months. Although there are many duct burners in operation in California, finding a site that was interested in demonstrating a new duct burner technology required considerable effort. Locating the site proved to be difficult for several reasons, with the most important being:

- For CHP applications, the duct burner is typically specified and installed at the time of initial system installation. Retrofit opportunities are very limited.
- Once a CHP system is on-line, the systems typically operate with almost no down time. Finding a site willing to be shut down long enough to allow installation and testing proved to be difficult.
- Timing of site availability had to match the authors' project schedule.

With assistance from Southern California Gas Company, two field demonstration sites were identified; both were in the South Coast AQMD and were industrial customers of the Gas Company. A review of both sites was conducted to see if they met the selection criteria.

- The target application was an industrial turbine in the 3-5 MW size range that operated for a sufficient number of hours to allow for significant life test data to be collected.
- The site would have already been designed to operate with duct burners, and most likely would currently have duct burners installed but would be interested in replacing their current burners.
- The system layout would be compatible with the research team's hardware design with respect to duct dimensions, fuel supply pressure, and accessibility.
- The site would be interested in the efficiency and NO<sub>x</sub> benefits that could be provided by the ALZETA supplemental burner relative to the performance of its current hardware.
- The burner could be installed with minimal permitting modifications. As a technology demonstration, it would be desirable to install and operate without subjecting the demonstration site to any permitting modifications that would extend beyond the length of the research team's demonstration.

Using these selection criteria as a guideline, Cenveo Anderson Lithograph in Commerce, California, was selected as the demonstration site. Reasons for selecting Cenveo included:

- Their 3.7 MW Allison 501 engine (now Rolls Royce) is in the industrial-size range that is of primary market interest to the authors and is operated year-round by the customer.
- The customer currently uses duct burners and wanted to increase the heat input from their duct burner system. The HRSG steam output is used to drive a 1.2 MW steam turbine that generates additional power, and low-pressure steam from the system is used to drive two chillers.
- The duct burner assembly at the site is flanged in place in the duct, allowing for easy removal of the existing burners and installation of the ALZETA burners,
- Permitting did not appear to be a problem as evidenced by proactive work done by Cenveo to coordinate with the South Coast AQMD.

A comparison of the two sites identified by SCG and evaluated as potential demonstration sites is shown in Table 6.

**Table 6. Comparison of site hardware at Cenveo and alternate field location**

Parameter	Cenveo	Alternate
Prime Mover	Rolls Royce 501	Solar Centaur
Capacity (MW)	3.75	3.3
Existing Duct Burners	Forney (Davis Engineering)	Forney (Davis Engineering)
Duct Burner Capacity (MMbtu/hr)	16.0	25.0
Fresh Air Operating Mode	Yes	No
Primary Heat Recovery Device	Heat Recovery Steam Generator	Heat Recovery Steam Generator

A site visit was scheduled and provided the opportunity for representatives from ALZETA, Cenveo, the Energy Commission, and Sempra to meet. Sempra representatives attended the meeting because they had facilitated the initial introduction between Cenveo and ALZETA and because Sempra became a partner in the funding of the field demonstration.

### **3.1.1. Cenveo Facility Description**

As stated above, the gas turbine generates 3.7 MW of electricity. With no duct firing the steam turbine generates an additional 500 kW. With duct burners at low fire, 150 kW of steam turbine capacity is added bringing steam turbine output to 650 kW. At the time of the authors' visit, the currently installed duct burner could operate only at low fire for emissions reasons. This meant that the plant could not generate the remaining 550 kW of electricity that the steam turbine was capable of generating unless the current system was repaired or a new duct burner system was installed. Hence, this site seemed ideal based on the selection criteria.

In addition, the operating data were used to perform an energy balance around the CHP system components. Cenveo dimensions and operating conditions are shown to compare favorably to the ALZETA pilot scale test hardware in Table 7.

**Table 7. Comparison of Cenveo conditions and pilot-scale test conditions**

<b>Property</b>	<b>Cenveo</b>	<b>ALZETA Pilot-Scale</b>
<b>Duct Internal Width (ft)</b>	7	1
<b>Duct Internal Height (ft)</b>	4.5	1
<b>Duct Area (ft<sup>2</sup>)</b>	31.5	1
<b>Duct Section Length (ft)</b>	2	1.2
<b>TEG Mass Flow (pph)</b>	122,378	3,000
<b>TEG Temperature (°F)</b>	1053	800
<b>Vol. Flow (scfs)</b>	459.4	18
<b>Vol. Flow (acfs)</b>	1336.6	43.6
<b>Bulk Velocity (scfs)</b>	14.6	18
<b>Bulk Velocity (acfs)</b>	42.4	43.6
<b>Heat Input (MMBTU/hr)</b>	12.5	1.2 – 2.0
<b>Burner Surface Area (ft<sup>2</sup>)</b>	20	2
<b>Burner Projected Area (ft<sup>2</sup>)</b>	10	1
<b>Duct Velocity (ft/s)</b>	21.2	21.8
<b>Face Velocity (ft/s)</b>	10.6	10.9

### **3.1.2. Test Schedule and Requirements for Field Access**

Concerning the scheduling difficulties that are involved, Cenveo shuts down only two times per year for scheduled service: Memorial Day weekend and Thanksgiving weekend. At the time of the first meetings with Cenveo, the opportunity to install over Memorial Day, 2007, had already passed, so Thanksgiving of 2007 was chosen as the installation date.

A second necessary decision was whether to build a new duct section or modify the existing duct. The original plan called for building a new duct section to replace the existing section so that the burners could be preassembled. However, after several visits to the demonstration site, it was observed that the initial plan to remove the existing two-foot duct section would most likely be more difficult than expected. The duct section at Cenveo has been in place for more than 10 years, and it is likely that some metal deformation has occurred due to high temperature cycling of the system. It is also possible that this deformation is significant, and the existing duct will become distorted when removed. If this were to occur, it may not be possible to install the new duct and, in a worst case scenario, impossible to reinstall the original duct.

Since installation had to be completed in one weekend, the worst case scenario for replacing the duct was that, at the end of the weekend, the turbine exhaust duct might still be open. The customer would then be unable to run the gas turbine, the steam turbine, or the supplemental burner system. This would be unacceptable and too great of a financial risk for the customer. However, if selecting to modify the existing duct and install burners in the field, the worst

case scenario is that the supplemental burner system for some reason might not be operational, but the gas turbine and steam turbine would both be running.

## **3.2. Hardware Design**

Unlike the pilot scale tests, the system for Cenveo had additional constraints for interfacing with the site—controls had to interface with the existing turbine and steam system, and the burner system was to be designed for retrofit. Work focused on interfacing with existing controls and the existing duct section.

### **3.2.1. Control System**

During the initial visits to the Cenveo site, the existing duct burner control hardware was inspected several times. It was assumed that significant modifications would be required, and possibly that the duct burner control panel would have to be replaced.

Discussions with the control panel designer determined that modifying the existing system would be costly and complicated. The existing panel is built around a programmable logic controller (PLC), which in general would provide flexibility in design. Unfortunately, due to the age of the PLC in the panel (built in the mid-1990s), a complete replacement of the PLC, IO boards, and other electronic hardware would probably be required. This would have been unfortunate, since the existing controls worked well for the existing system.

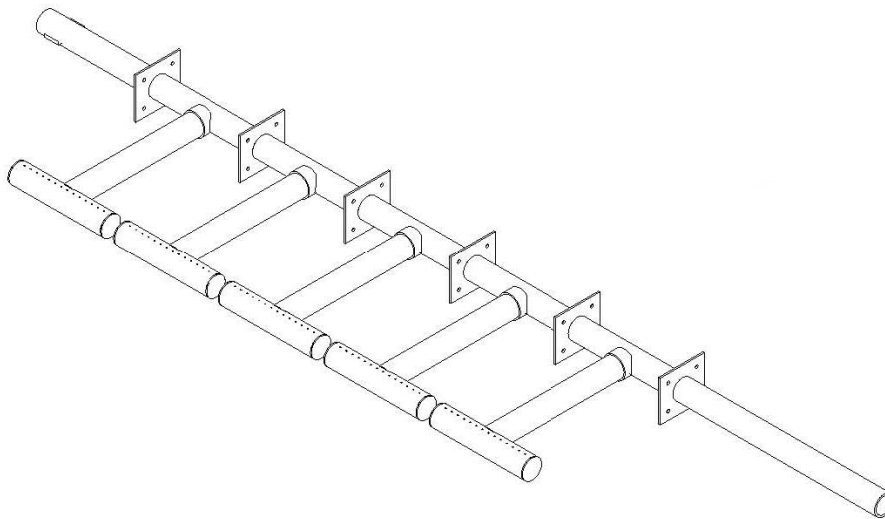
By limiting the existing controls to system on-off control, the flame safeguards, system “permissives” functions, and the majority of the existing controls could be reused for the ultra-low NO<sub>x</sub> burner system. The existing panel was left intact, but one function, the “heat input” signal, which is the signal from the control room to the main gas valve of the existing duct burner, was moved to a new panel. With this approach, schedule and budget risks were both reduced.

The existing system used a 4-20 mA control signal to modulate an actuator that controlled the main gas valve. This control approach would not have worked with the ALZETA burner concept. As discussed previously, the ALZETA concept requires that each module in the system operate in an on-off mode. A separate control panel was designed by ALZETA that intercepted the 4-20 mA gas valve control signal and ran it through a logic circuit that broke the 4-20 mA analog signal into discrete on-off commands for each of the 10 ALZETA modules. These discrete on-off commands then actuated an individual fuel solenoid valve for each of the 10 ALZETA modules, thereby matching the heat input requirement to number of modules being operated.

Choosing this controls approach simplified the interface requirements for the site. The only modification to the existing control panel was the redirection of the 4-20 mA heat input signal. In addition to being simpler and less expensive to implement at the Cenveo test site, this design also demonstrated that minimal changes to what is considered to be “standard duct burner controls” would be required in future ALZETA duct burner installations.

### 3.2.2. 10 Module Design

The field system module design is very similar to that of the final pilot-scale module design. However, at the field demonstration site the modules have to be supported by the gas pipe and cannot be flanged in place as with the pilot-scale tests. Because the gas pipe needs to pass through the modules close to the burner face, it cannot be used for gas injection as there would not be enough length for the gas and air to mix fully. To overcome this issue, the burner module was lengthened relative to the pilot-scale design, and an extension pipe that projects back to the entrance of the burner module was installed. Attached to each of these pipes is an “injection pipe” located in the center of the module and spanning across the horizontal length of the module. Figure 15 shows the gas injection design. Modeling of the gas injection was done using Lefebvre’s theory on mixing jets in cross flow, as it was with the pilot-scale mixer (Lefebvre, p.117-126.) The injection pipe has 14 holes, each 1/8” in diameter, that inject the gas at a 90° angle into the air stream facing straight up and straight down.



**Figure 15. Gas injection design.**

The next step in the duct burner design was the arrangement and support of the 10 modules. The final design had the duct burner modules arranged into two rows, or banks, that span five modules across without any space between modules. The modules were placed side by side to ensure that each module would light the next. The two rows were centered both horizontally and vertically within the duct, leaving only a six-inch space between the two rows. The six-inch space was left to ensure that one row of burners could light the second without the flames impinging upon one another. The burner banks were centered within the duct to ensure they receive enough air and to give the most even air distribution between the two banks possible.

### 3.3. ALZETA In-House Testing

To test fire the newly designed burner system before installation at the Cenveo site, an existing ALZETA high-capacity air blower and burner control system were incorporated into a full-scale test facility. The test facility consisted of a blower mounted to a frame, an expansion duct section, and a duct burner section. A variable-frequency drive (VFD) was used to control the blower motor speed to provide air flow up to 20,000 cfm. The expansion section increased the duct area from the 2 ft x 2 ft blower exit up to the 7 ft x 4.5 ft duct dimensions that match the Cenveo site. Photos of the test facility hardware follow in Figures 16 through 19.



Figure 16. Side view of the expansion section.



**Figure 17. Front view of the expansion section.**



**Figure 18. Blower mounted on frame.**



**Figure 19. Duct section with one burner bank installed.**

The tests at ALZETA would accomplish the following:

- Demonstrate reliable pilot ignition of the duct burners to ensure the pilot could light the burners in a high cross-flow velocity environment. There was some concern that the velocity in the duct may affect pilot performance.
- Demonstrate that the first burner bank could be used to ignite the second. The baseline design had the burners spaced six inches apart. This test would determine whether that distance was sufficiently close to have reliable ignition from the first bank of burners, or if a second ignition pilot would need to be installed. It also would determine whether the burners were separated enough to avoid the flames from one bank impinging on the other.

### **3.3.1. Full-Scale Duct Testing at ALZETA**

Testing began with one completed burner row (five burner modules) installed on the duct section. Of the five burners, the two on the side closest to the pilot lit and were operating with uniform combustion, but the other three never had a steady flame. The cause of this was determined to be poor air flow distribution across the duct. Using a hot-wire anemometer, the air velocity in the duct was tested for uniformity. It was discovered that the velocity on the right side of the duct, which did not light, was 50-60 ft/s while the velocity on the left side of the duct was only 5-10 ft/s. The extra air on the right side caused the premix to be too lean to light.

To make air distribution more uniform, the entire area around the burners was blocked with 51% open area perforated metal. The velocity profile in the duct was measured again using an anemometer, and the air distribution had the same pattern, but with less severe velocity variation from point to point. The right side of the duct had velocities in the 10-13 ft/s range, and the left side had velocities in the 1-4 ft/s range. The burners were lit again, but with the same results as the first test.

Realizing that the air distribution across the duct could not be fixed with the current blower, tests were done to even the air distribution through the burner modules. To accomplish this, the anemometer was used to measure the velocity on the face of each burner module. Sheets of perforated metal were placed at the entrance of each burner module to restrict the flow. An even distribution occurred when the first burner on the left had no restriction, the second was covered with 65.5% open perforated metal, and the last three were all covered with 35.4% open perforated metal. The burners were lit again, and all remained operating with an even flame distribution.

A second burner bank was received and installed as shown in Figure 20. Initial tests again resulted in poor flow distribution and combustion. A more thorough analysis of flow was performed. There were several points of concern:

- Single module tests completed earlier in the project met the research team's emissions and thermal performance goals. The 10-module 2-bank system was designed as a scale-up of the single module hardware, with the 10 individual full-scale modules being nearly identical to the single-module test hardware. This represents the same geometric configuration needed at Cenveo. While the pilot facility was not capable of matching the temperature within the Cenveo duct, the mass flow at ambient temperature was the same and would allow matching the burner surface firing rate. It was believed that the previous single-module test results performed at temperature would also allow scaling the cold flow results to TEG conditions, but this was not the case.
- It could be assumed that the poor flow uniformity and poor flame stability observed during the full-scale testing at ALZETA would also be a problem in the field demonstration, as the actual localized flow conditions in the duct at the field site were unknown. There was no practical way of measuring how flow deviates spatially across the Cenveo duct and how it fluctuates as a function of time.



**Figure 20. Field demonstration burners installed in ALZETA test facility.**

### **3.3.2. Modifications**

#### ***Test Duct Modifications***

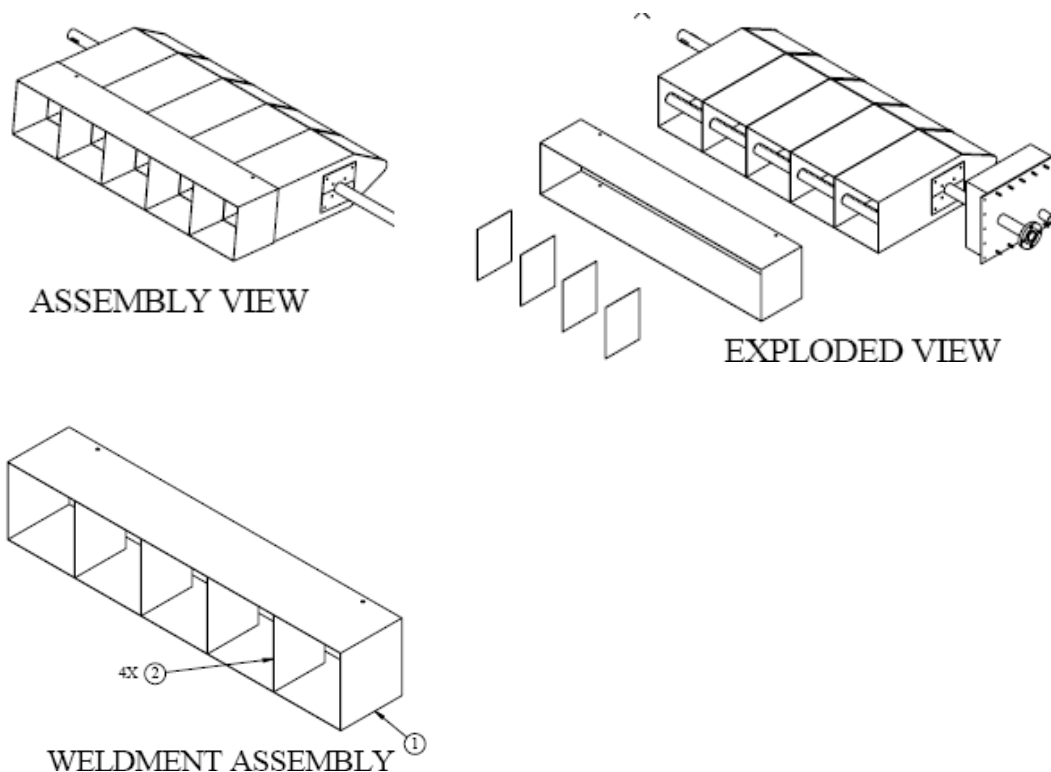
Initial modification work was directed toward improving flow conditions in the duct. As initially designed, the ALZETA test duct ended immediately downstream of the burner surfaces, while in the field installation the flow continues in a duct downstream to the HRSG. It was thought that flow disturbances at the duct exit could be caused by the rapid flow expansion from the duct to open air or by wind, and that these disturbances could be causing the flow non-uniformity. Neither of these flow conditions exists at the test site geometry, so the research team's test facility was modified to more closely simulate the field configuration.

First, a straight-walled constant cross-section duct was built to simulate the duct section at the job site. This duct is 8 feet long, which provided adequate downstream flow uniformity. Test results were disappointing as the duct extension had minimal impact on improving burner uniformity. Next, honeycomb-type flow baffles were added to the transition duct section that exists between the blower and burners. The baffles were added to remove large-scale flow disturbances from the blower wheel from propagating downstream. Test results following these modifications also showed minimal improvement in performance. Based on these results it was determined that, although the flow in the duct is not steady or uniform, improving flow conditions in the duct did not significantly improve burner performance.

### ***Burner Mixing Modifications***

Other work focused on improving mixing in the individual burner modules. Changes were made to the gas injection spuds, the modules were extended to allow for more mixing length prior to the burner surface, and internal and external perforated metal baffles were added to improve mixing. Of all these modifications, the most significant were determined to be:

- Burner modules were increased in length by 10 inches by adding an “extension box” to the upstream end of the burner modules, as shown in Figure 21. This box is made from 14 gauge stainless steel, which was similar to the material used for the burner housings. This modification added approximately 50% to the mixing length of the burner modules. During development the extension box was bolted to the burner modules. In the field installation, the extension box was to be welded to the burner modules after they were installed.



**Figure 21. Schematic of extension box that was added to the burner modules to improve mixing of fuel with turbine exhaust gas.**

- The gas injector design was changed to incorporate more gas jets and less flow area between banks of jets. This modification approximately doubled the number of jets per module from 28 to 56. The gas spuds for the new injector design are shown in Figures 22 and 23. The old spuds were similar in design but had fewer holes and larger hole diameter.

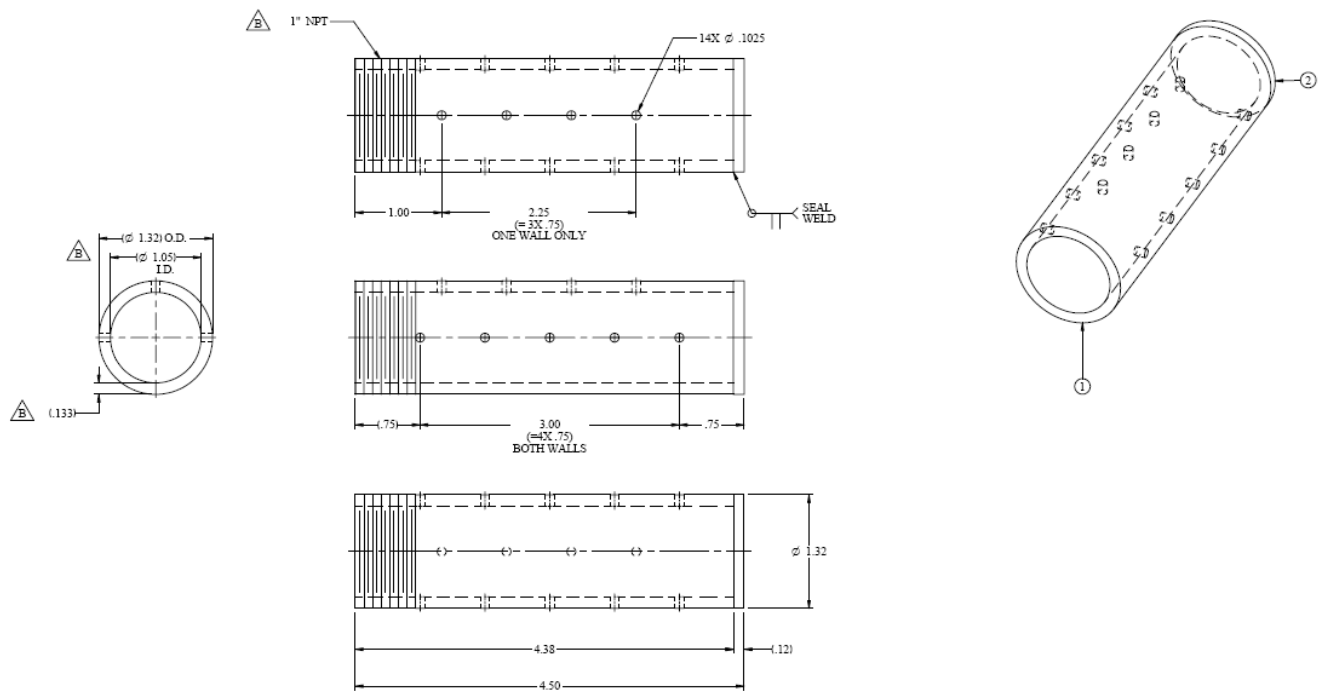


Figure 22. Modified gas injectors for use in "H" design mixing arrangement.



Figure 23. Assembled modules, extension box, and gas injectors.

### **3.4. 2007 Thanksgiving Weekend Burner Installation**

Installation was scheduled for Thanksgiving weekend, 2007, when Cenveo would be performing regular maintenance on the turbine system. As part of the installation plan, work in the duct had to be completed by Sunday morning so that the turbine could be restarted.

Specific activities to be completed and the schedule for completing these activities were as follows:

- Thursday Nov 22 – ALZETA and subcontractor RECON (the onsite contractor) entered the turbine exhaust duct, removed the existing duct burner, modified the duct walls to accommodate new penetrations for the ALZETA burner, and began insulation and sheathing repair as required.
- Friday Nov 23 – The ALZETA auxiliary control panel for fuel flow control was installed by Cenveo personnel, Figure 24. ALZETA and RECON completed insulation and sheathing work. ALZETA started installation of the 10 burner modules and the pilot and scanner mounting plate as shown in Figures 25 and 26.
- Saturday Nov 24 – ALZETA completed installation of burner modules and supervised the installation of the perforated steel baffles around the burners. Pilots were installed.
- Sunday Nov 25 – Turbine was scheduled to go on-line at 10 AM. All “in duct” work by ALZETA had been completed the previous day. During turbine startup, ALZETA and RECON mounted the duct burner gas train. ALZETA personnel completed the fuel piping to the individual burner modules. ALZETA gas train work was completed Sunday afternoon, as shown in Figures 27 and 28.

The installation proceeded essentially on schedule, with no significant problems encountered. Photos of the installation are shown on the following pages.

### **3.5. Startup**

The plant operators restarted the gas turbine as scheduled at 10 AM Sunday, Nov. 25. As stated earlier in this report, ALZETA had completed all in-duct work that was scheduled. However, there remained approximately 8 hours of work to do outside of the duct including piping the gas manifold, and installing the gas pilots, and the flame scanner mounting hardware. This work was completed during the day.

At about 5 p.m. on Sunday, a visual inspection of the burner modules via peepsites installed on the duct revealed that one burner module (top row, 4th burner from wall) had suffered some significant structural damage. The top plate of the burner module had separated from the rest of the box and had moved forward about 6 inches. The burner surface material was still attached to the burner, but since the premix enclosure that makes up the burner module had been breached, this module could no longer be operated. This first damage was observed approximately 7 hours after the turbine was started. At the time, it was believed that only one module had failed, and the test would proceed with lightoff of the other modules later in the week.

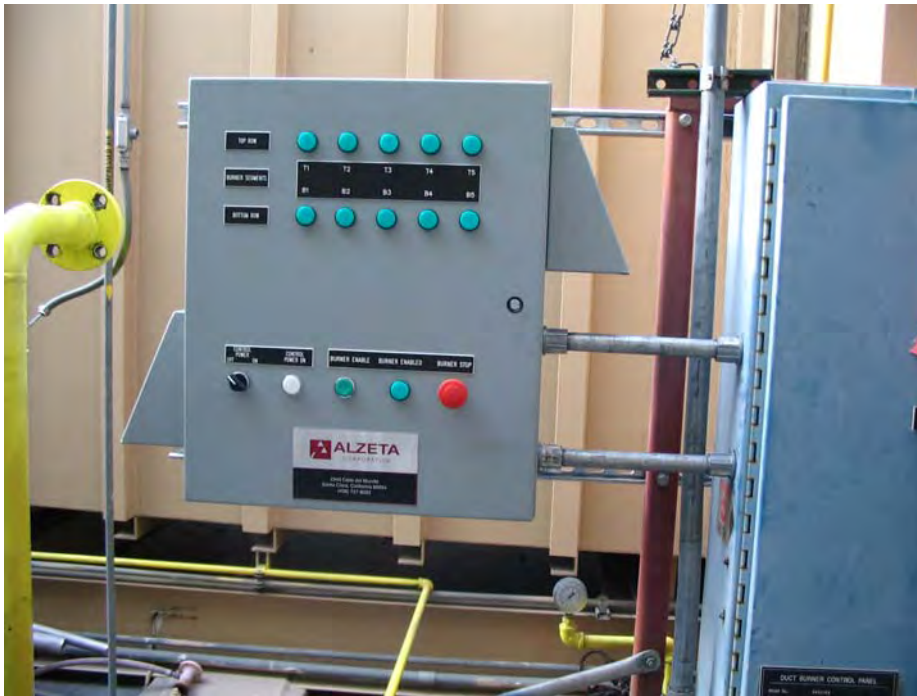


Figure 24. ALZETA auxiliary panel for controlling of fuel valves for individual burner modules.



Figure 25. Front view of completed assembly of burner modules.



**Figure 26. Final installation of perforated steel plate at bottom of burner modules.**



**Figure 27. View of gas manifold, control valves, burner mounting plates, and pilot plate.**



**Figure 28. View of far side of duct. The modifications required the addition of the two mounting posts at the center of the photo.**

The plant operators monitored the modules starting on Monday, and by Tuesday morning a second module had failed. Since visual access is limited, the condition of all the burner modules was uncertain. Since two modules had now failed, there was increasing concern over the potential of causing damage to the exhaust duct or steam generator. Hence, the plant facility operators decided to shut the turbine down on Tuesday evening and remove the duct burner modules the next morning.

ALZETA was at the jobsite for the removal of burner modules on Wednesday morning. The burner hardware was inspected inside of the duct prior to any removal activities. But in order to minimize downtime, all modules were removed by cutting with a torch. Additional damage was sustained beyond what occurred during startup and operation. Before removal, it was observed that damage had occurred to approximately half of the burner modules. The decision to shut down the turbine and remove modules was the correct decision.

Photos of the damaged burners are shown in Figures 29 through 31. The most apparent problem observed during removal was that a significant number of welds had completely failed. There was no evidence of metal fatigue nor was there any sign of distortion of the metal parts around the welds. This was particularly true for the extension boxes attached to the inlet end of the burner modules.

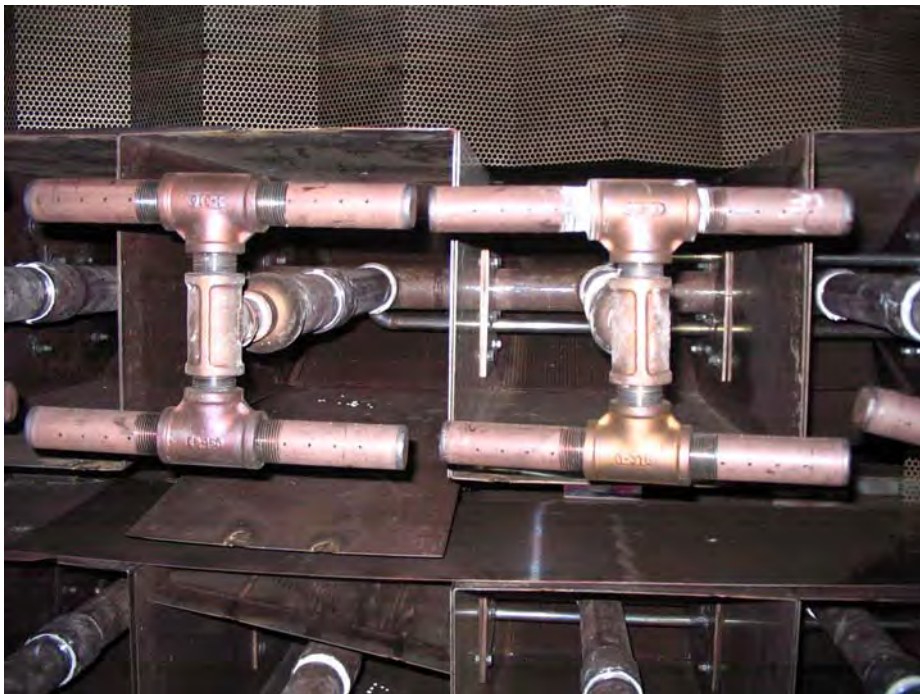


**Figure 29. Example of damaged burner module. Damage was caused by top side of burner module breaking loose and tearing the burner face material.**





**Figure 30. Photo of undamaged burner module taken at same time as photo in Figure 27. Approximately half of the modules were undamaged.**



**Figure 31. View of rear of duct burner. Note that the top of the right-hand module is missing and the bottom of the left hand module has nearly separated. Also note that the extension box is not in the photo because it is no longer attached to any of the burner modules.**

The failure of the duct burner hardware after such a short time in the duct was a major disappointment and a setback to the project. Parts were shipped from the field site back to ALZETA for evaluation.

### **3.6. Post-Test Analysis**

#### **3.6.1. ALZETA Testing of Welds**

Damaged hardware was returned to ALZETA and inspected. The key defect was in welds joining various sheet metal parts. The fabrication procedures, weld processes, and installation processes were reviewed with the research team's subcontractors. The weld procedures were insufficient for the duct burner service environment. The weld procedure was optimized to limit distortion of the parts rather than ensure structural stability.

A more rugged hardware design had to be developed. Building everything with heavier gauge (thickness) material would allow for more substantial welds at the expense of system weight and ease of installation. Increasing module weight would also require a redesign of the burner support system. The module and system weights as a function of thickness are summarized in Table 8:

**Table 8. Weight as a function of material gauge**

<b>Material Thickness</b>	<b>Module Weight</b>	<b>Approx. System Weight</b>
<b>Gauge (inches)</b>	<b>lbs</b>	<b>lbs</b>
14 GA (0.0747)	33.1	500
10 GA (0.1345)	59.6	900
8 GA (0.1644)	72.8	1100
3/16 plate (0.1875)	83.1	1300

The existing design was reviewed, and several alternatives were developed. Some further development of the candidate designs is required to establish their impact on system performance, installation interferences, and cost.

#### **3.6.2. Review of Materials of Construction**

Also analyzed were the hardware design and method of fabrication. Certain aspects of the design must be maintained in order to achieve the research team's performance objectives. Specifically:

- The design has to include a "premix section," which requires that an enclosed duct with sufficient mixing length be an integral part of the design.
- Within the premix section, a fuel injection system is required that efficiently distributes and mixes the fuel and air.
- At the combustion end of the premix section, the design has to incorporate a porous metal surface for the flame attachment.

Some design constraints are imposed by the need to meet these objectives. The original design met these objectives, but was not sufficiently rugged. Hence, future construction will include

thicker gauge sheet metal parts and thicker and more penetrating welds. Also, metal reinforcement (such as gussets) may be added to critical areas.

### **3.6.3. Liftech Analysis**

Liftech Consultants, Inc. of Oakland, California, a company with previous experience in the structural analysis of duct burners, was hired to conduct a structural review of the design. Two initial areas for improvement were identified and evaluated:

1. The 7-foot span of the main burner support is relatively long and could be subject to significant deflection and vibration. This suggests using intermediate beam supports between the duct walls to support the burner assembly.
2. The duct baffles that were fabricated from perforated steel were attached to the duct walls and to the burner modules. It is preferable to support the baffles only from the duct walls and to isolate them from the burner modules.

The Liftech analysis also concluded:

- Stresses on the individual burner modules due to dead load and static TEG flow loads were well within allowable limits. The gauge of metal used and method of construction were most likely not the cause of the failure. The general design approach used in the first demonstration can be used, but some modifications should be implemented.
- Since localized failures were observed along seam welds, an alternate method of burner module fabrication was recommended. This method should provide more weld area and a reduction in stress at the weld. This change was implemented and tested in the research team's field demonstration.
- Since dead load and static flow loads were well within allowable limits, failure could have been caused by what Liftech referred to as "flutter." This refers to movement of the boxes possibly due to unsteady loading caused by flow oscillations and could also be due to vibrations at resonant frequencies. In order to deal with the flutter problem, it was recommended that the back end of the burner boxes be separately supported to eliminate the cantilever that exists with the current design. It was also recommended that a larger diameter, and therefore stiffer, main support pipe for the modules be used.
- It was recommended that the design be modified to include a space between each module to allow for thermal expansion. With no allowance for expansion, stresses can be introduced to the modules.
- There was concern that high temperature creep (a permanent plastic deformation) of the main burner support pipe could occur over time. This could lead to stress on the individual burner boxes as the support pipe would sag due to creep. The initial field test did not last long enough for creep to occur. However, it was recommended that this problem be addressed in future demonstrations by using a stiffer support pipe and the addition of the space between individual modules in each row of burners.
- The distribution grid should be mounted to the duct walls and not to the individual burner modules. Space for thermal expansion should exist between the grid and the modules.

### 3.7. Project Rescope

Following failure analysis and initiation of a redesign of the basic duct burner components, ALZETA wanted to retest the new hardware at another field site prior to the end of the project. Cenveo was ruled out as a site for the retest. The limited access to the burner hardware, except during plant shutdowns over Thanksgiving and the Memorial Day weekends, made it difficult for researchers to commit to a “one-shot” redesign and retest of the hardware.

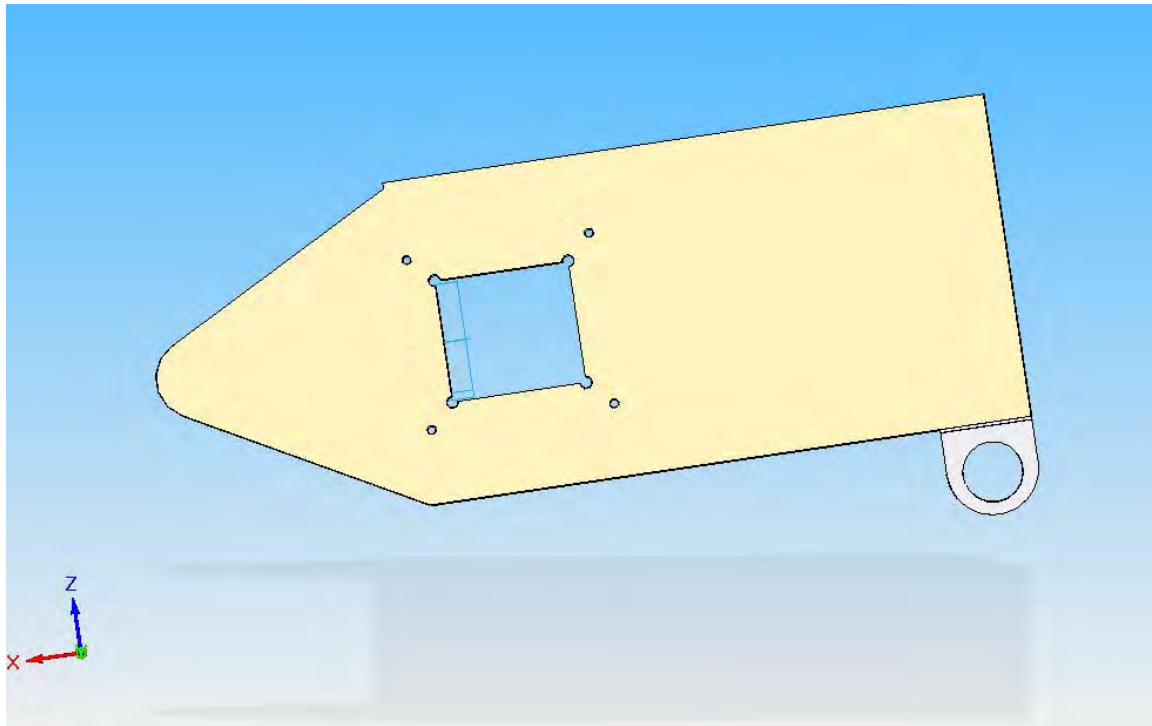
Other lower risk field test sites were identified and contacted. Of these, most promising were Stelter & Brinck and an HRSG manufacturer, both of which ALZETA had worked with in the past. Stelter & Brinck was the only company that could provide a location to test within the authors’ project time constraints. Hence an agreement between Stelter & Brinck and ALZETA was prepared to allow immediate implementation and testing of the suggested redesigns. Testing was then performed at Stelter & Brinck at typical environmental conditions for an industrial duct burner application.

Stelter & Brinck provided the additional advantage of also being interested in pursuing a commercial agreement with ALZETA at the end of the demonstration. It expressed an interest in selling the burner into commercial hot air ducts as well as into gas turbines exhaust ducts. The design and testing of the modified duct burner hardware are discussed in the following section.

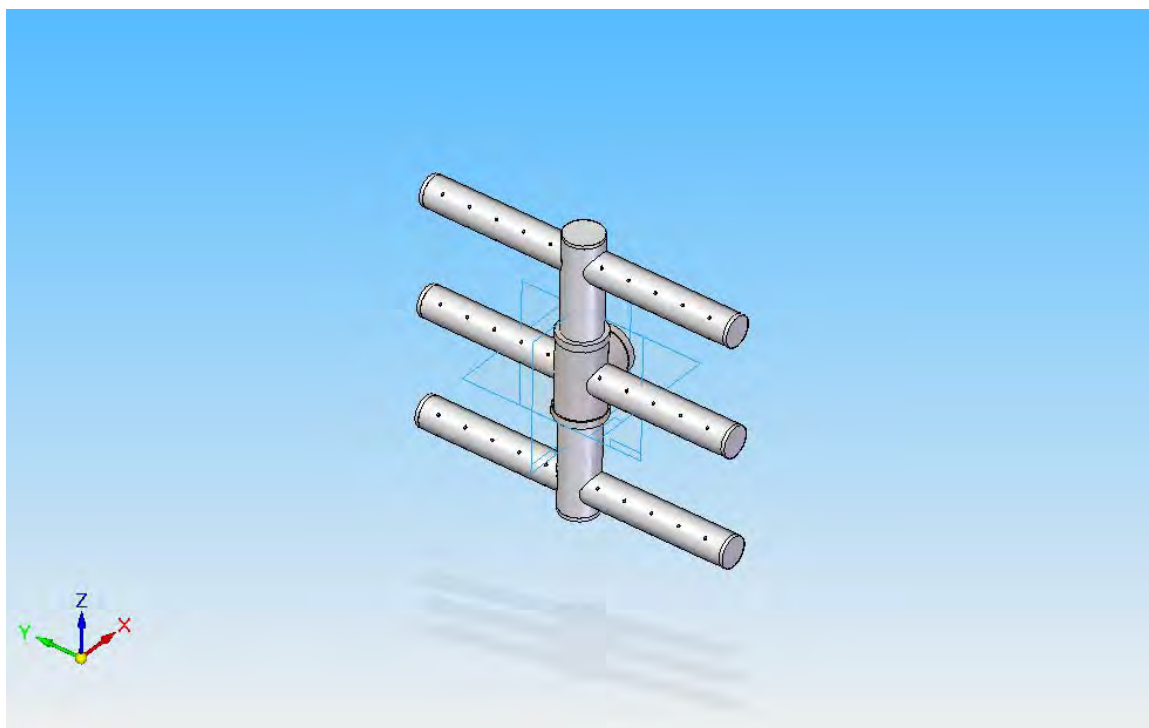
### 3.8. Design Modifications

Based on Liftech analysis, a new burner module was designed and constructed to include the following features:

- Add support to the back end of the module to eliminate the “cantilever” and stabilize the module vertically. This would eliminate any “flutter” and unsteady loading due to flow oscillations. A bracket was added to the rear of each module for connecting to a support pipe that installs in the duct, as shown in Figure 32.
- The mounting arrangement between modules was modified to allow spacing for thermal expansion.
- Modules would be mounted on a stiffer and larger diameter support pipe. In addition, stress relief radiuses were included at the corners of the square feed-through hole that accommodates the support pipe. These are also shown in Figure 32.
- The flow distribution grid was mounded directly inside the duct and not supported by the burner modules.
- A new multiple injector pipe design (Figure 33) was developed to improve mixing and eliminate the need for the extension box installed at the Cenveo site. This eliminates much of the mechanical loading experienced at Cenveo.



**Figure 32. New burner box manifold with bracket for rear support rod pipe.**



**Figure 33. Three branched injector tee.**

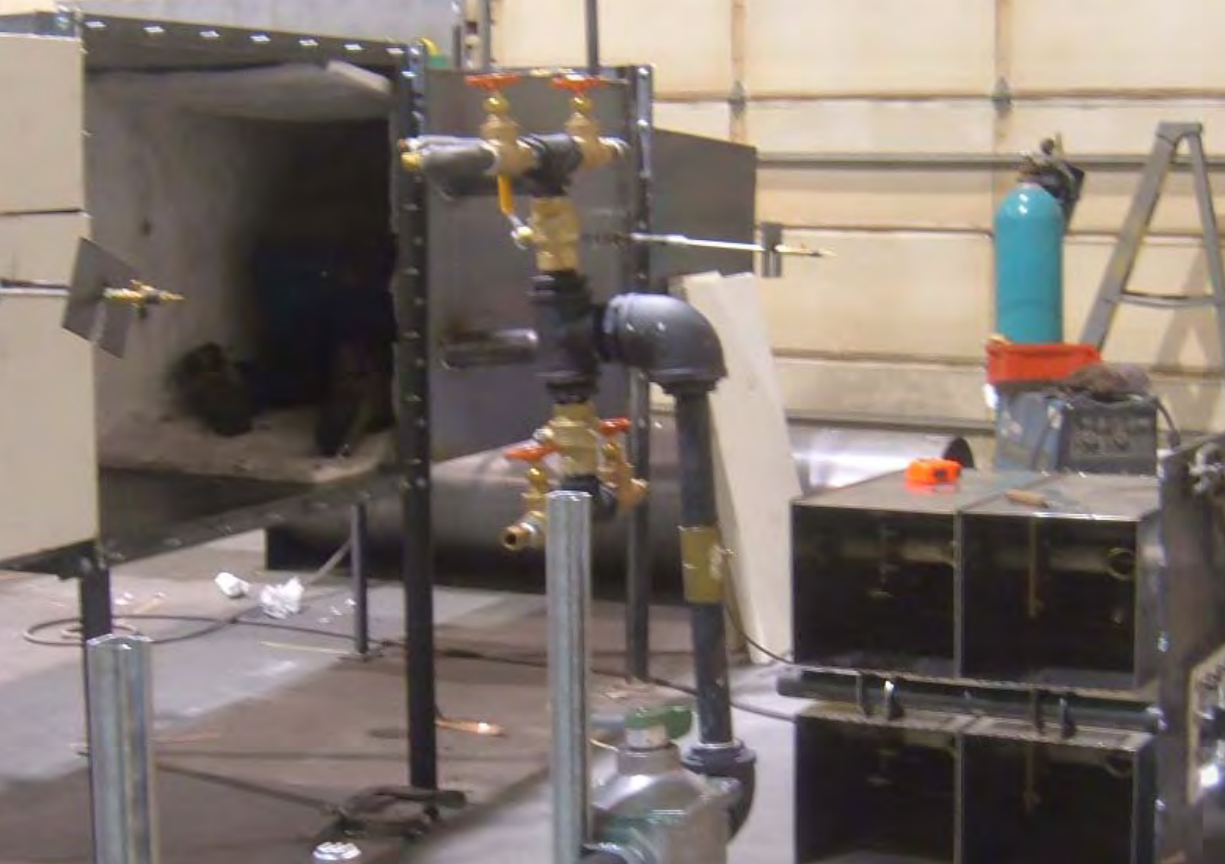
### 3.9. Stelter & Brinck Duct Test Facility

A test duct at Stelter & Brinck in Cincinnati was used to fully simulate either a gas turbine or industrial recirculation process heater. The burner that provided the exhaust gas and heat to the system, or turbine exhaust generator (TEG), was mounted upstream of the duct burners by about 16 feet and provided up to 800°F of oxygen depleted air (down to about 16% O<sub>2</sub>) at velocities typical of TEG applications. A recirculation system, which brought flue gas back to the first duct section, also allowed the O<sub>2</sub> and duct velocity to be optimized for any given test. The facility is shown in Figure 34.



**Figure 34. Test facility at Stelter & Brinck.**

The facility included a burner section with removable wall to provide full access to the duct burners so that any modifications could be easily implemented. Also included were numerous ports for measuring temperature, velocity, emissions, etc. The open section, with the burner modules ready for installation, is shown in Figure 35.



**Figure 35. Burner bank to be installed in duct.**

### **3.10. Test Results**

Commissioning activities were performed with inlet temperatures to the duct burners at 500°F. Both the upper and lower row of burners were easily lit by the ignition pilots. The 500°F temperature was chosen both as a moderate temperature condition to check out system operation, as well as representative of a level that meets the low end of the air heater duct burner operation requirements.

The first series of tests were also conducted at 500°F, with the TEG operating at about 15 percent excess air and 125 ppm NO<sub>x</sub> and 300 ppm CO. Once both rows of burners were lit, a jet of flame could be seen between the two rows of burners, as shown in Figure 36.



**Figure 36. Flames occurring in the gap between burners during testing.**

The burners were then taken out and inspected. There was no evidence of damage, or a spot where the gas might be escaping from the mixing manifold boxes into the flow stream. Despite the fact that the 2.38 inch OD support pipe was blocking the 3.75 inch wide gap between the burners, it was thought that this gap could be channeling and enhancing the duct gas flow through this gap. If so, a high-velocity stream through this gap could entrain local gases at higher static pressure, such as may reside at the entrance of the burner modules. This could result in some fuel being entrained into the duct flow, which could result in flaming between the burners. To address this possibility, and further block this gap to the duct flow, a strip of 30 percent open area perforated plate was inserted across this gap. The injector jets on the top and bottom rows of the three pronged injector “tree” were also turned 180 degrees so that the 0.10-inch diameter injection ports faced downstream, instead of upstream, into the flow. Lastly, a second, eddy-dampening, 40 percent open area perforated plate was installed in the upstream duct section, 20 inches in front of the burners, to damp any large-scale eddy formation.

With the above changes made, the system was restarted. There appeared to be less of a jet between the burners, but there was still a flame. Some measurements were made at 500°F inlet temperatures, and then the system was taken up to 700°F for more thorough testing including measurements of pressure drop and emissions.

For the next series of tests, a second, 30% open area perforated plate strip was overlaid on the first one, also attached at the gap between the burners. The double row “H style” injectors, which had been used previously at the Cenveo site, were installed. For additional assurance that fuel would not be entrained or diverted out of the mixing manifold boxes, the placement of the injector tees was moved downstream ½” into the burner box. This new configuration then

matched the configuration design in the previous installation, and no fuel bypassing was expected. However, upon restart, some flaming was still apparent between the rows of duct burners.

In an attempt to both diagnose the problem as well as operate at different system conditions, the research team decided to reduce the dilution air damper setting to reduce overall system flows. The system was stopped and restarted, such that a pilot measurement of velocity in the duct was reduced to 0.45 inches w.c. from the original reading setting of 0.88 inches w.c. When the burners were relit, flaming between the burner modules was still present. The flames were different in that they appeared more yellow than before. Out of concerns about possible flashback, the system was shut off before emission data measurements were taken.

The burner was once again removed, and a 3.5-inch-wide, solid plate was attached at the gap between the burners, at the inlet plane of the burner boxes. Also, the “H injector” tree “branches” were turned 180 degrees so that all gas jet ports were facing either perpendicular to the flow, or downstream, and none facing upstream. The dilution air damper setting was returned to its original position for the final testing in the afternoon. These final modifications appeared to eliminate the flame in the gap, and testing proceeded. At 500°F duct gas temperature, a premix reading in the first burner module in the lower row was measured at 6% O<sub>2</sub> (wet) in the burner box. This corresponds to about 45% excess air or dilution condition in the module upstream of the burner surface.

The inlet temperature to the duct burners was then taken up to 800°F. Two data points were taken, the first with 17.6% O<sub>2</sub> in the upstream duct, and 14.2% O<sub>2</sub> in the exit stream. Another premix measurement was taken at this point and indicated 7.5-8% O<sub>2</sub>, wet, which corresponds to approximately 65% dilution in the burner. Measured input to the burners was 3.3 MMBtu/hr, and the clocked gas flow was 8 MMBtu/hr total input to the TEG simulating burner and duct burners.

For the final test, the flue gas recirculation (FGR) damper on the dilution air line was opened to allow some recirculation of flue gas products, and further reduce O<sub>2</sub> levels in the duct. Final measurements, upstream of the burner, were 17.6% O<sub>2</sub> in the duct, and 15.1% O<sub>2</sub> in the exit. The measured first burner gas firing rate was 2.5 MMBtu/hr, and the measured duct burner gas flow was about 2.3 MMBtu/hr. A summary of test measurements is shown in Table 9.

**Table 9. Summary of test results**

Test #	Preheat T (°F)	Number of Burners	Duct %O <sub>2</sub>	NO <sub>x</sub> ppm 15% O <sub>2</sub>	CO ppm 15% O <sub>2</sub>	Duct Burner dP in. w.c.	Duct Burner SFR MMBtu/hr-ft <sup>2</sup>
1	500	Two	19.4	2.3	444	0.7	0.55
2	700	Four	18.6	6.6	791	0.9	0.46
3	500	Two	19.3	3.8	506	0.6	0.58
4	500	Four	19.3	6.3	721	0.7	0.48
5	800	Four	17.9	9.2	588	0.9	0.41
6	800	Four	17.6	4.3	444	0.6	0.29

To reduce the NO<sub>x</sub> and CO data to their reported forms (ppm, dry, corrected to 15 percent O<sub>2</sub>), raw, uncorrected dry emissions data was taken for the first, or turbine exhaust gas generator (TEG) burner, in the duct upstream of the burners. These emissions were then corrected, on a dry volume basis, to the new volume downstream of the burners after the duct burner fuel gas had been injected. These adjusted emission levels were then subtracted from the NO<sub>x</sub> and CO emissions that were measured downstream of the burners. This difference between the adjusted upstream emissions and the downstream emissions was attributed to the duct burners. Using the percentage O<sub>2</sub> in the exit chamber, these emissions difference figures were corrected to 15 percent O<sub>2</sub>.

### **Conclusions**

Except for Test 1, in which only one row was operating, burner NO<sub>x</sub> emissions did not meet the sub-3 ppm project goals. Furthermore, in all tests, CO levels were very high. The source of the high CO production was not clear, and there was no time to investigate CO production mitigation measures.

Operation of the burners appears to be sensitive to upstream pressure and flow conditions. If they are fluctuating, burner operation will be fluctuating, also. This mode of operation can produce high CO levels, related to both mixing and to the flame going on and off intermittently, or attaching and unattaching, on the burner pad surface. If the pressure field on the burner surface is constantly changing or moving around with a fluctuating flame, the upstream pressures and flows will also be constantly changing. This lack of steady state conditions has deleterious effects on both mixing and on flame stabilization, making proper operation of the burner a challenge. It is possible that forcing more flow through the burners would improve operation, with the limit being all flow passes through the burners and there is no bypass flow. The options for this type of operation would need to be assessed.

Regarding the flames occurring outside the burners, the previous explanation of those phenomena is the best explanation at this time. Because this is not fully understood, a final design would need provisions for blocking and balancing flow between the burner modules.

In spite of difficulties encountered with combustion stability and emissions, structural integrity of the redesigned hardware was demonstrated. Testing at Stelter & Brinck verified the redesigned burner assemblies would withstand the duct environment with no structural issues. Ignition and combustion controls, which were never tested at Cenveo, did perform well.

## **4.0 Conclusions**

This project developed and demonstrated an ultra-low NO<sub>x</sub> supplemental firing burner system for reheating turbine exhaust gas prior to a heat recovery steam generator. Exhaust gas heating is desired in cogeneration applications in that it permits more effective heat recovery from the prime mover (turbine or engine) exhaust. The higher gas temperature also provides greater flexibility, as the amount of steam produced in the HRSG can be adjusted independently of the gas turbine power demand. This allows separate control of facility electric and steam (heat) outputs. However, as supplemental firing is applied in California, these burners must also meet the strict NO<sub>x</sub> emissions standards being implemented throughout the state.

### **4.1. Achievements**

At both laboratory and pilot scale, in which a full-scale supplemental firing burner element was operated at typical TEG conditions, NO<sub>x</sub> and CO emissions of sub-3 ppm (corrected to 15% O<sub>2</sub>) were achieved. In addition, the selected wedge-shaped configuration offered greater than 2:1 burner surface area to duct cross-section area, matching the typical incoming heat flux requirements for a HRSG. Hence conceptually, the ultra-low NO<sub>x</sub> supplemental firing duct burner technology developed under this project met the goals of emissions, heat output, and operating flexibility for plant operators seeking higher overall efficiency and more independent control over steam (heat) and electric loads.

### **4.2. Turbine Exhaust Gas Environment**

Within simulated or actual turbine exhaust gas environments, however, performance was disappointing. Structural failure occurred at Cenveo where the full-scale burner assembly was subjected to actual turbine exhaust conditions. In addition, under full-scale simulated TEG conditions, combustion stability on the burner surface was not always achieved, resulting in high emissions and poor burner turndown. A much greater understanding of the turbine exhaust gas environment is needed, as well as much more testing under actual conditions once this environment is better defined.

### **4.3. Industrial Hot Air Environment**

This environment is similar to TEG in that the incoming stream is at elevated temperatures and somewhat depleted of O<sub>2</sub>. Again, at pilot scale, the supplemental firing burner module demonstrated it could meet objectives when subjected to this environment. However, at full-scale, the turbulent conditions resulted in poor combustion stability. But in contrast to the gas turbine environment, many industrial hot air heating applications can tolerate additional pressure drop, using outside combustion air, or withdrawing process air from the duct and externally premixing with fuel. This would eliminate the issues with mixing with using injectors installed in the turbulent duct environment, allowing banks of full-scale burner modules to operate much more like the conditions experienced in the pilot-scale tests. Implementing external premixing would be straightforward using readily available equipment.

#### **4.4. Sales Potential in the Industrial Hot Air Market**

Stelter & Brinck reports a strong demand for ultra-low NO<sub>x</sub> supplemental firing burners in the process industries. In addition to the need to meet more stringent emissions requirements, NO<sub>x</sub> also can affect the quality of products in the food industry and others. With the structural design proven at Stelter & Brinck, and previous emissions and turndown performance proven during pilot testing, the product could proceed to the market assuming external premixing of the fuel and air could be implemented.

#### **4.5. Sales Potential in the Gas Turbine Exhaust Market**

The cogeneration market seeks maximum efficiency and minimum pressure loss downstream of the turbine. It is, therefore, more difficult to justify adding external premixing of the fuel and air. Hence, additional testing of in-duct mixing techniques, as well as a more detailed understanding of the performance tradeoffs associated with baffles and other pressure-consuming means of stabilizing duct flow, needs to be well understood before a product can be finalized for this market.

#### **4.6. Recommendations**

Although the testing at Stelter & Brinck verified that the redesigned modular burner system was structurally sound, the turbulence and flow variations within the TEG environment prevented stable combustion and low emissions. Providing a high-efficiency ultra-low NO<sub>x</sub> burner system for CHP facilities will require the following, and these are the authors' recommendations for future development activities:

- Develop a greater understanding of the TEG environment and how to make flow more uniform within ductwork. The research team attempted to minimize flow variations by installing perforated metal plates to increase flow resistance (a common means of improving flow distribution). Yet, flow in a typical TEG duct was still too turbulent for this approach to be effective.
- Use slipstream combustion air and eliminate in-duct air-fuel premixing. Pilot scale demonstrated that ultra-low NO<sub>x</sub> could be achieved when air and fuel flow rates were stable. This is not the case in a turbulent TEG duct—the air flow inherently varies widely. However, withdrawing TEG from the duct, externally mixing it with fuel, and piping the premixed TEG and fuel back to the burner modules would be very similar to the pilot scale test situation. The authors believe developing a design for this approach (which has not been done but should be straightforward) would lead to more rapid and successful commercialization.

## Glossary

Specific terms and acronyms used throughout this work statement are defined as follows:

Acronym	Definition
acfm	actual cubic feet per minute
AFT	Adiabatic flame temperature
Btu	British thermal unit
cfm	Cubic feet per minute
CHP	Combined heat and power
CPR	Critical project review
CSB	ALZETA commercial surface-stabilized burner product
Dynamic Pressure	Density multiplied by velocity squared divided by two $g_c$ $(\rho \cdot v^2)/2g_c$
FGR	Flue gas recirculation
$g_c$	Gravitational constant. 32.2 (lb <sub>m</sub> -ft)/(lb <sub>f</sub> -s <sup>2</sup> )
HRSG	Heat recovery steam generator
in. w.c.	Inches of water column
LNB	Low-NO <sub>x</sub> burner (typically 20-30 ppm NO <sub>x</sub> )
MBtu	1 thousand British thermal units
MMBtu	1 million British thermal units
MW <sub>e</sub>	Megawatts electric power
PIER	Public Interest Energy Research
ppm	parts per million (usually on a volumetric basis)
psig	Pounds per square inch, gauge
scfm	standard cubic feet per minute
SFR	Surface firing rate of burner (Typical units are MMBtu/hr/ft <sup>2</sup> )
TEG	Turbine exhaust gas
UCC.1	Uniform Commercial Code (Financing Statement)
ULNB	Ultra-low NO <sub>x</sub> burner (typically 15 ppm NO <sub>x</sub> or lower)
VFD	Variable-frequency drive



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